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U. S. DEPARTMENT OF AGRICULTURE

Design and Operation of Small Irrigation Pumping Plants

Carl Rohwer, Irrigation Engineer, Soil Conservation Service

UNITED STATES DEPARTMENT OF AGRICULTURE
WASHINGTON, D. C., OCTOBER 1943



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Design and Operation of Small Irrigation Pumping Plants¹

CONTENTS

By CARL ROHWER, *irrigation engineer, Division of Irrigation,
Soil Conservation Service*

| | Page | | Page |
|---|------|---|------|
| Introduction..... | 1 | Fitting equipment to conditions—Con. | |
| Water requirements of area to be irrigated..... | 3 | Piping and auxiliary equipment..... | 44 |
| Capacity of well or surface water supply..... | 3 | Bids and purchase agreements..... | 61 |
| Fitting equipment to conditions..... | 6 | Testing pumps, engines, and motors..... | 63 |
| Pumps..... | 6 | Cost of pumping for irrigation..... | 69 |
| Motive power..... | 22 | Summary..... | 74 |
| Drives..... | 37 | Literature cited..... | 78 |

INTRODUCTION

It is not the purpose of this circular to recommend the construction of new pumping plants for irrigation during the present emergency. Their construction at this time should be given consideration only when the plants will increase the production of essential foods or other materials vital to the prosecution of the war. Many plants now in use, however, supply water necessary for these purposes. Some of them are obsolete or will wear out in the near future. Many of them are inefficient. It is important that these plants be kept in service at high efficiency by proper replacements or repairs, thereby increasing production and reducing the use of power. To do this requires a thorough knowledge of the principles of pumping-plant design and operation. Since this information is not available from other sources it is published here to help in keeping indispensable plants in operation at high efficiency.

¹ This circular was prepared under a cooperative agreement with the Colorado Agricultural Experiment Station, under the direction of W. W. McLaughlin, Chief of the Division of Irrigation, Soil Conservation Service. The information contained herein is based on observations of pumping plants; interviews with operators in California, Oregon, Arizona, Idaho, Colorado, Nebraska, and Kansas; and on data compiled from publications of State agricultural experiment stations and equipment manufacturing companies. Acknowledgment is made of assistance from W. E. Code of the Colorado Experiment Station; members of the Division of Irrigation; and the manufacturers, dealers, and individuals who supplied photographs for illustrations or information for the report.

Equipment and materials required for pumping plant maintenance, repairs, and replacements are rationed by the War Production Board, but these supplies can be obtained under a preferential allocation of material for agricultural purposes when the need for them is clearly shown. The plant owner, however, should find out from the manufacturer or jobber whether the materials needed are available before starting any major repairs or replacements.

Many farmers who plan to install pumping plants for irrigation lack the technical training needed in designing such plants, and consequently depend on representatives of the extension service, experiment station, or other State or Federal agency for advice regarding the appropriate type and size of pump, power unit, and accessories. In many instances, especially in States where irrigation pumping is not practiced extensively, these advisors are not irrigation engineers or pump specialists, and they likewise need technical information to pass along to intending irrigators. It is the purpose of this circular to supply such information. Although prepared primarily for those who advise the farmer, the information it contains will be useful also to farmers who wish to extend their knowledge of pumping. Other bulletins of the United States Department of Agriculture discuss the subjects of irrigation wells and the feasibility of pumping for irrigation (15, 16).²

The importance of irrigation to crop production, which is of such vital concern to the Nation at the present time is clearly set forth in table 1. This table shows the total acreage of crops in the 11 Western States, the acreage irrigated, and the total crop value as well as the value of irrigated crops.

TABLE 1.—*Total value and acreage in crops and acres irrigated in 11 Western States*

| State | Cropland harvested, 1939 ¹ | | | Value, 1939 ² | | |
|-----------------|---------------------------------------|-----------------|---|---|-----------------|--------------------------------------|
| | Total, all cropland | Irrigated crops | | Total, all principal crops ¹ | Irrigated crops | |
| | | Total | Proportion of irrigated to all cropland harvested | | Total | Proportion of irrigated to all crops |
| | <i>Acres</i> | <i>Acres</i> | <i>Percent</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Percent</i> |
| Arizona..... | 525,974 | 471,372 | 89.62 | 25,318,000 | 22,334,000 | 88.21 |
| California..... | 6,534,562 | 3,732,215 | 57.11 | 340,521,000 | 280,120,000 | 82.26 |
| Colorado..... | 4,769,671 | 2,136,754 | 44.80 | 59,889,000 | 46,927,000 | 78.36 |
| Idaho..... | 2,935,350 | 1,578,741 | 53.78 | 64,523,000 | 43,970,000 | 68.15 |
| Montana..... | 5,748,069 | 1,359,126 | 23.64 | 51,135,000 | 18,882,000 | 36.93 |
| Nevada..... | 435,855 | 430,743 | 98.83 | 5,044,000 | 4,885,000 | 96.85 |
| New Mexico..... | 1,572,507 | 409,351 | 26.03 | 23,322,000 | 13,223,000 | 56.70 |
| Oregon..... | 2,824,316 | 730,682 | 25.87 | 66,892,000 | 19,302,000 | 28.86 |
| Utah..... | 966,088 | 761,093 | 78.78 | 21,729,000 | 19,037,000 | 87.61 |
| Washington..... | 3,569,803 | 409,161 | 11.46 | 94,275,000 | 36,382,000 | 38.59 |
| Wyoming..... | 1,534,800 | 1,019,653 | 66.44 | 17,736,000 | 14,531,000 | 81.93 |
| Total..... | 31,416,995 | 13,038,891 | 41.50 | 770,384,000 | 519,593,000 | 67.45 |

¹ As tabulated in the Census of Agriculture, 1940.

² As estimated by Paul A. Ewing, Irrigation Economist, Soil Conservation Service, on basis of statistics in the Irrigation Census, 1940.

² Italic numbers in parentheses refer to Literature Cited, p. 78.

WATER REQUIREMENTS OF AREA TO BE IRRIGATED

The quantity of water that must be supplied by the pumping plant is the difference between the total water requirement of the crop and the rainfall, plus the quantity necessary to take care of losses. These losses may vary between 10 and 50 percent of the water applied, depending on character of soil, length and kind of distribution system, and the care exercised in applying the water. If some water is available from other sources, such as streams, canals, or reservoirs, the quantity to be pumped may be reduced by the amount obtained from those other sources. The total quantity of water required ranges from 18 to 42 acre-inches per acre. Of this quantity from 4 to 6 acre-inches is probably the minimum that will have to be supplied by pumping and 42 inches the maximum, except in special cases. Complete information about water requirements of crops is given in bulletins (6, 7, 8, 9) of the Division of Irrigation, Soil Conservation Service.

In most localities the demand for water reaches a peak in June or July. The pumping plant must have sufficient capacity to supply that portion of the peak demand not available from other sources. The requirement for the month of highest use will range from 20 to 40 percent of the total quantity required by the crops for the entire season. Areas where the growing season is short have the highest peak demand. For short periods the maximum water requirements may exceed the capacity of the pump, but if the pump is designed to supply water enough to meet the needs during the month of highest use the crops will probably not be injured seriously by the temporary shortage unless the farm is small or there is little diversity in crops.

CAPACITY OF WELL OR SURFACE WATER SUPPLY

The area that can be irrigated by the pumping plant will depend on the capacity of the well or the surface water supply from which the water is pumped. If the water is lifted from a reservoir that holds sufficient water, the rate of pumping can be adjusted to fit the needs of the crops, but if the water is obtained from a stream or well, the rate will be limited by the capacity of the well or to the quantity from the stream to which the farmer is entitled. The laws governing the appropriation of water from streams and reservoirs in most of the Western States are definitely established, and it is important that the farmer comply with these laws. However, since not all the Western States have laws covering pumping from ground water, there may be some uncertainty as to the right to the water from this source.

The peak demand for water by crops usually occurs when streams are low, and, as many streams are already overappropriated, it is possible that all the water will be taken by those users who have senior rights. Where pumping for irrigation is extensive there is generally a gradual lowering of the water table (static water level in artesian areas) during the irrigation season and partial or complete recovery during the remainder of the year. Consequently, the lift on pumps in wells often is greatest at the time of peak demand. If this greater lift can be overcome by lowering or speeding up the pump or by other means any temporary shortage in supply can be made up. However, if the water level regularly fails to recover during the nonpumping

season an overdraft on the ground-water storage is indicated, which may result in so lowering the water level as to make pumping economically infeasible.

The economic limit of lift varies widely in different parts of the country and in different parts of the same area, depending on the cost of fuel or power, the crops grown, the climate, the productiveness of the soil, and the ability of the irrigator. The average lift and the estimated economic limit of lift for pumping plants in each of the Western States are set out in another bulletin (15). Because the lift of a specific plant does not exceed the values given in this report, it does not necessarily follow that the project will be a success. Likewise, because a plant exceeds the limit it will not necessarily be a failure. So many factors enter into the success or failure of a plant that even though the lift be unfavorable other factors may overbalance this disadvantage. There are many areas in which the recommended lift is being exceeded, apparently with success. However, in those cases where it is necessary to exceed the recommended lift, the other factors involved should be carefully investigated to make sure that their effects are counterbalancing.

The discharge of a well is a function of the draw-down, by which is meant the distance the water level is lowered while the pump is in operation. This relationship is ascertained by a well test made by pumping at various rates and noting the draw-down for each different discharge (fig. 1). A pumping test should always be made after completion of a well in order that the necessary data will be available when purchase of the pump is undertaken.

As illustrated by the diagram (fig. 1) yield may be expected to increase with draw-down, but yield is increased at the expense of increase in lift. In water-bearing formations of small thickness the increase in yield per foot of draw-down decreases as the draw-down increases. Usually it does not pay to draw a well down to the level of its maximum capacity because the water level will reach a point where the increase in draw-down is greater proportionately than the increase in yield. If water is particularly valuable, it may be desirable to pump the well to a point where the yield per foot of draw-down decreases rapidly, but in no case should the water be drawn down continuously beyond the economic limit of lift in order to increase the supply.

If the well is in a thick water-bearing formation or an artesian formation, the discharge is very nearly proportional to the draw-down so long as it does not exceed the artesian pressure head (fig. 1). In wells of this type it is customary to increase the draw-down until the required capacity is obtained, but as before, the draw-down should not be increased until the pumping lift exceeds the economic limit.

All the water pumped from the well must be lifted from the level to which it is lowered by the draw-down due to pumping. Not only does the water gained by increasing the draw-down have to be raised

from this level, but also all the water obtained from the previous level must now be lifted from the lower level. It is apparent, therefore, that careful consideration should be given to the balance between the increase in cost of pumping and the value of the water gained. Considering only the cost of the well and pumping equipment and of power for pumping, the most economical capacity (or draw-down) is ascertainable by estimating the total number of acre-feet that may be used during the season and then computing the total cost per acre-foot including fixed charges on the well and equipment and power cost for different discharge rates. The fixed charges per acre-foot will

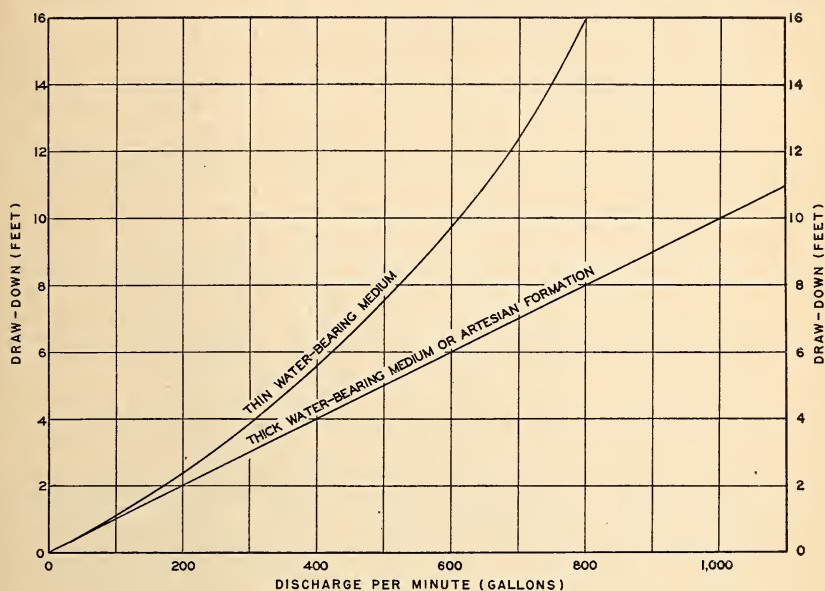


FIGURE 1.—Typical discharge draw-down relation of a well in a thin water-bearing formation and one in a thick water-bearing formation or an artesian stratum.

decrease as the quantity of water pumped increases; whereas, the power cost will increase as the draw-down increases. At some point the sum of these costs will be a minimum. Other considerations, such as the size of stream needed for efficient irrigation, or the number of hours daily during which it is possible or convenient to carry on irrigation, may require a larger discharge than the most economical rate shown by these computations. If the desired discharge is much greater than the economical rate, it may be wise to use a second well either supplying the same pump or entirely independent. (See U. S. Department of Agriculture Circular 546, Putting Down and Developing Wells for Irrigation, for information regarding battery wells.)

FITTING EQUIPMENT TO CONDITIONS

PUMPS

In choosing a pump for a well where the static water level is 20 feet or more beneath the surface, first consideration should be given to a deep-well turbine. If the water level is closer to the surface than 20 feet, a horizontal centrifugal pump will be cheaper and will probably have a longer life. Either a single-suction or a double-suction horizontal centrifugal pump may be used, the principal advantage of the double-suction pump being that the impeller can be inspected by opening the case without disturbing the piping. For pumping from streams and reservoirs the horizontal centrifugal is probably most satisfactory. However, if the water level fluctuates widely, a vertical centrifugal or a propeller-type pump may be better. A propeller pump will give best results for low lifts, but when a pump of this type is used sufficient power must be available for pumping against the maximum head. Approximate prices of horizontal single-suction centrifugal pumps appear in table 2.

TABLE 2.—*Cost of horizontal, single-suction, ball-bearing, centrifugal pumps of different sizes, typical capacities, and required operative horsepower¹*

| Size (inches) | Dis- charge per minute | Theo- retical power required, per foot of lift | Pump effi- ciency | Actual power required, per foot of lift ² | Pump with pulley | | Pump with base ³ | |
|------------------|---------------------------------|---|-------------------------|--|---------------------|----------------|-----------------------------|----------------|
| | | | | | Weight | Cost | Weight | Cost |
| | <i>Gallons</i> | <i>Horse- power</i> | <i>Percent</i> | <i>Horse- power</i> | <i>Pounds</i> | <i>Dollars</i> | <i>Pounds</i> | <i>Dollars</i> |
| 2..... | 200 | 0.051 | 40-60 | 0.13 | 172 | 55 | 232 | 95 |
| 3..... | 300 | .076 | 65-75 | .12 | 261 | 70 | 336 | 116 |
| 4..... | 500 | .126 | 65-75 | .19 | 284 | 100 | 360 | 147 |
| 5..... | 700 | .177 | 65-75 | .27 | 436 | 120 | 536 | 173 |
| 6..... | 1,000 | .252 | 70-80 | .36 | 462 | 140 | 562 | 193 |
| 8..... | 1,500 | .379 | 70-80 | .54 | 533 | 200 | 583 | 253 |

¹ The prices are f. o. b. warehouse and are for the year 1940.

² Efficiencies are taken as the lower values in the preceding column.

³ Includes flexible coupling for direct-connected motor, but does not include motor.

Although the horizontal centrifugal is cheaper and has a longer life than the deep-well turbine, the latter type is often chosen for well installations even where the standing water level is less than 20 feet beneath the surface, because it does not require priming nor is a pit necessary in which to install it. The deep-well turbine is also replacing the vertical centrifugal. Furthermore, there is a notice-

able tendency for farmers to buy the better grade deep-well turbines which are designed to fit their individual conditions and consequently are usually more efficient. The prices of deep-well turbines for various conditions are given in table 3. Screw or propeller pumps are best adapted to operation under conditions where the lift is under 10 feet, but they may be used when the lift is from 20 to 30 feet if the necessary stages are added. For lifts less than 10 feet, the efficiency of screw pumps is much higher than that of the horizontal centrifugals. While screw pumps usually are made in sizes too large for small plants, some companies make them in the smaller sizes.

TABLE 3.—*Installed prices of oil- or water-lubricated deep-well turbine pumps with different types of drives*

| Capacity (gallons per minute) | Set- ting | Bowl diam- eter ¹ | Stages ¹ | Col- umn dia- meter ¹ | Motor size ¹ | Installed price ² of pump with— | | | |
|----------------------------------|--------------|------------------------------------|---------------------|---|----------------------------|--|-------------------|-------------------|-------------------|
| | | | | | | Direct drive | Belt head | | Gear head |
| | | | | | | 1,760 r. p. m. | 1,160 r. p. m. | 1,760 r. p. m. | 1,760 r. p. m. |
| | <i>Feet</i> | <i>Inches</i> | <i>Number</i> | <i>Inches</i> | <i>Horse- power</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> |
| 225----- | 30 | 7½ | 1 | 5 | 3 | 475 | 405 | 350 | 495 |
| 225----- | 50 | 7½ | 2 | 5 | 5 | 600 | 500 | 430 | 600 |
| 225----- | 80 | 7½ | 3 | 5 | 7½ | 760 | 665 | 550 | 765 |
| 450----- | 30 | 9½ | 1 | 6 | 5 | 560 | 425 | 390 | 560 |
| 450----- | 50 | 9½ | 2 | 6 | 10 | 785 | 575 | 510 | 670 |
| 450----- | 80 | 9½ | 2 | 6 | 15 | 935 | 810 | 630 | 780 |
| 900----- | 30 | 9½ | 2 | 8 | 10 | 755 | 600 | 490 | 645 |
| 900----- | 50 | 12 | 1 | 8 | 15 | 880 | 770 | 575 | 735 |
| 900----- | 80 | 12 | 2 | 8 | 25 | 1,290 | 1,010 | 870 | 960 |

¹ Bowl diameters, number of stages, column diameter, and motor horsepower apply only to pumps with speed of 1,760 r. p. m.

² Prices are for the year 1940. They include motor, 10 feet of suction column, strainer, starter, and cost of wiring between transformers and motor for direct-connected motor-driven units, but do not include engine, belt, or universal shaft for units with other types of drives. The cost of double universal shaft is \$37.50 for sizes up to 10 hp. and \$50 for sizes from 10 to 25 hp. Pulleys suitable for combination belt and gear drives cost about \$15 each.

PUMP CHARACTERISTICS

The horizontal and vertical centrifugal and the true deep-well turbine operate on the centrifugal principle. The mixed-flow turbine is a combination of the centrifugal and the propeller types, in which the action of the pump is the result of a combined centrifugal force and direct thrust. In the axial-flow propeller or screw-type pump there is very little centrifugal action, the water being moved by the thrust of the blades of the propeller. Impellers of the different types are shown in figure 2 (*I*).

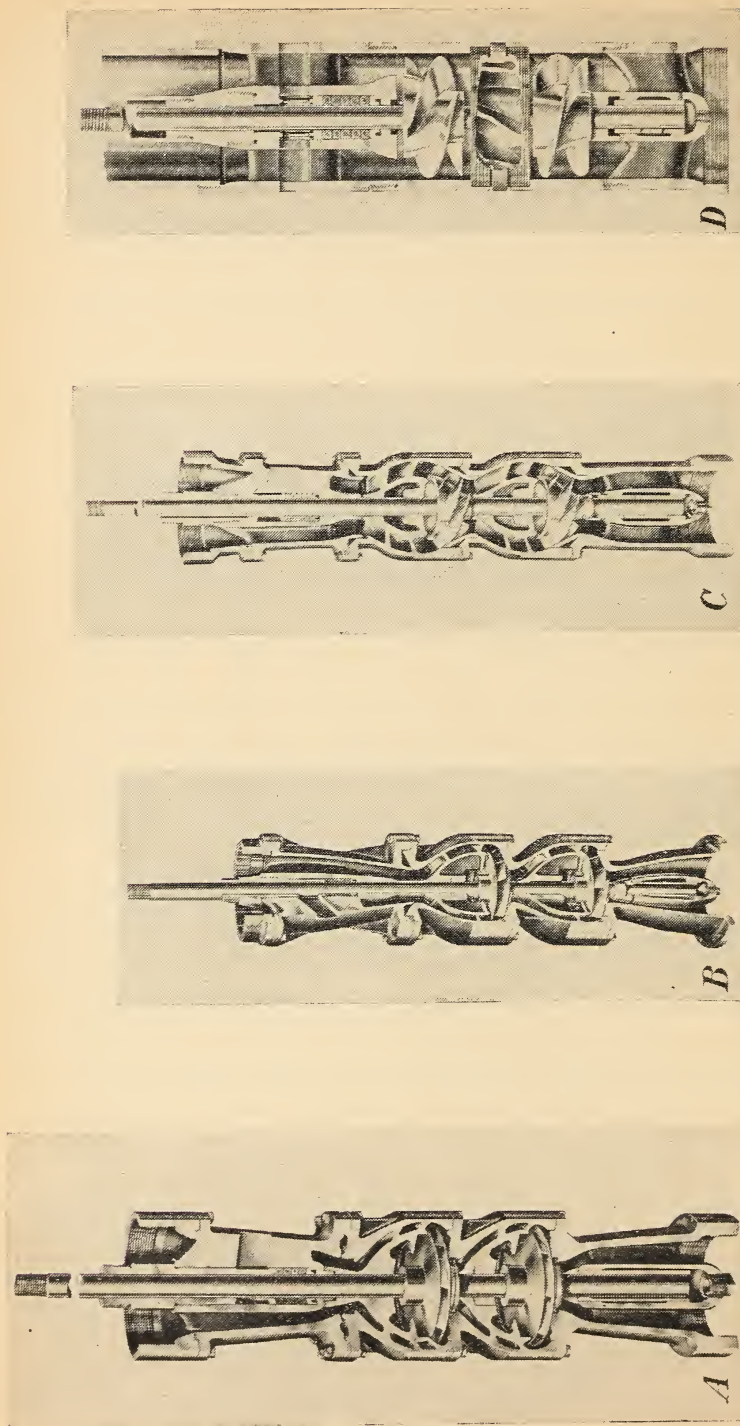


FIGURE 2.—Section of two-stage deep-well turbine bowls: *A*, With impellers of the centrifugal type; *B*, with semienclosed impellers of the centrifugal type; *C*, with mixed-flow impellers, a combination of axial-flow and centrifugal types; *D*, with impellers of the screw or propeller type.

Centrifugal-type impellers are either enclosed, semienclosed, or open. Enclosed and semienclosed impellers are used in horizontal and vertical centrifugals and deep-well turbines. Open impellers are found only in horizontal and vertical centrifugal pumps. Mixed-flow impellers are used only in deep-well turbines. They may be of either the enclosed or semienclosed types (figs. 2, *A* and *B*). Screw- or propeller-type impellers are without shrouds. They are similar to propellers on boats, and their action is the same.

Most horizontal centrifugal pumps for irrigation have semienclosed impellers; closed impellers are found mostly in the more expensive double-suction pumps such as are used in municipal pumping plants. They are also the most common type in deep-well turbines. Since the size of the impeller in deep-well turbines is definitely limited by the diameter of the well, it sometimes happens that it is not possible to obtain the full capacity from a good well of small diameter by a pump with impeller of the centrifugal type. An increase in capacity is obtained by the use of a mixed-flow impeller, which provides a more direct passage for the water and consequently makes it possible to obtain more water from a well of given diameter. If still greater capacity must be obtained, it can be done by the use of a pump with axial-flow impellers of the type shown in figure 2, *D*.

Mixed-flow impellers are usually made with semienclosed vanes, as shown in figures 2, *B* and *C*. The open side of the impeller is turned accurately to fit the seat in the bowl of the pump, which also is accurately machined. As the edges of the vanes wear there will be leakage between them and the seat in the bowl, but this leakage can be reduced by lowering the impellers until the original clearance is obtained. This adjustment provides a means of bringing the efficiency of the pump back to practically its original value. However, if sand is pumped for long periods, the seat in the bowl becomes roughened, and it is then necessary to remachine the seat before the impeller can be adjusted properly. The capacity of pumps with semienclosed impellers can be decreased by raising the impeller in the event that the flow of the well decreases. This adjustment causes a reduction in the efficiency of the pump, but it is not as great as it would be if the capacity were reduced by throttling the discharge.

Enclosed impellers of the centrifugal type have a circular skirt attached to the bottom shroud which fits into an annular space in the pump bowl called the sealing ring (fig. 2, *A*). Both these surfaces are accurately machined so as to obtain a close-running fit because the seal at this ring prevents the water from leaking from the discharge side of the impeller back to the suction side. Since the difference in pressure is large there is always some leakage and, if the water carries sand, the parts wear rapidly. This wearing results in a considerable part of the water leaking back through the sealing ring. When this occurs it is necessary to pull the pump to make the repairs, which consist of reboring the annular seat in the bowls and building up the skirt of the impeller by welding on new material. The skirt must be accurately machined to fit the new seat. When sand is especially troublesome stainless-steel or hard-bronze rings are fitted to the skirt for the purpose of reducing the wear. At this point a double seal is sometimes used. This consists of a

rubber ring set into the bowl so that it is just below the bottom edge of the skirt of the impeller. By lowering the impeller a small clearance can be maintained even though there is considerable wear caused by the sand. It is important that sufficient clearance be maintained so that the impeller does not rest on the rubber seal ring, lest there be considerable loss of power due to the friction and injury to the rubber ring.

Impellers of the centrifugal type produce a high head per stage, but the quantity pumped is small. This is the type of impeller used on high-head installations where the diameter of the well is large enough to accommodate an impeller of the required diameter. Mixed-flow impellers produce a medium head per stage and screw- or propeller-type impellers produce the smallest head per stage but the largest discharge. An intermediate type impeller has characteristics between the mixed-flow and the propeller-type impeller.

The performance curves of the different types of impellers are shown in figures 3, 4, 5, and 6. The head-capacity curve shows the relation between the head and the quantity pumped; the efficiency curve shows the ratio of the useful work done by the pump to the energy expended when operating at any point on the head-capacity curve; and the horsepower curve shows the power required to drive the pump when operating under conditions indicated (that is, as to discharge, head, and efficiency). It does not include engine or motor losses. The efficiencies shown are for multistage pumps. For single-stage pumps the efficiencies are somewhat lower.

The discharge is indicated at the bottom of the diagrams, the head on the left side and the horsepower and efficiency on the right. A vertical line through any discharge will intersect the head-capacity curve at the point that shows how high this particular pump will lift this quantity of water. The intersection with the efficiency curve is the efficiency with which the pump will operate under these conditions, and the intersection with the horsepower curve is the horsepower required. All the curves are based on a constant speed (as given in legends of figures 4, 5, and 6).

The diagrams illustrate the fact that as the head increases the capacity decreases until the point of zero flow is reached. The corresponding head is called the shut-off head. The slope of the head-capacity curve is a measure of the sensitiveness of the pump discharge to change in head. A steep curve indicates that considerable change in head has a relatively small effect on the discharge. This is the type of head-capacity curve to be desired in an irrigation pump because changes that would ordinarily occur in the water level would have small effect on the discharge.

The efficiency curve rises slowly as the discharge increases until it reaches a peak, and then drops abruptly. Obviously, the pump should be operated at a point as near the peak efficiency as possible. For irrigation service the peak of the efficiency curve should be flat so that small changes in the pumping level and discharge will not result in a marked reduction of efficiency. For this reason the efficiency rating at heads, or discharges 10 percent above and below the normal operating capacity, should be considered when a pump is being purchased.

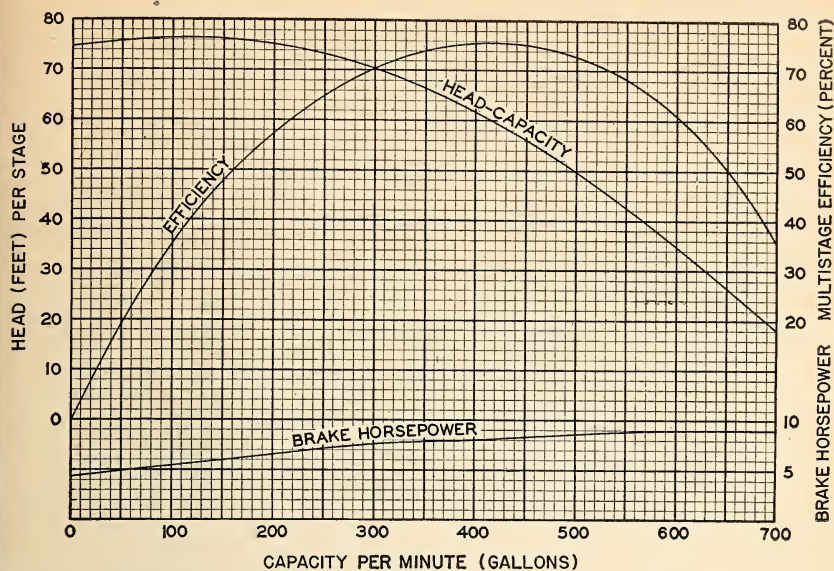


FIGURE 3.—Performance curves of deep-well turbine of the centrifugal type. The horsepower increases as the head decreases.

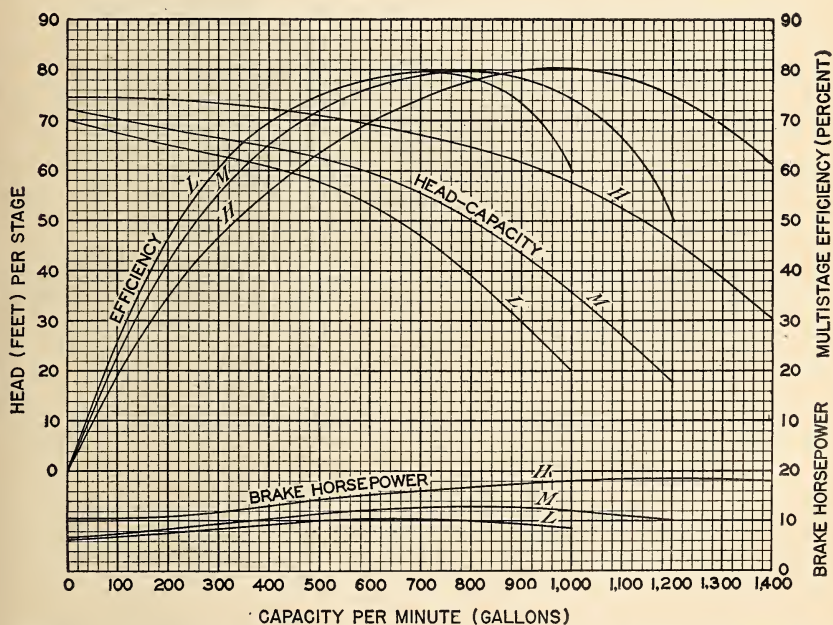


FIGURE 4.—Performance curves of deep-well turbine with special, semienclosed centrifugal impeller, speed 1,750 r. p. m. The horsepower is a maximum at approximately the peak of efficiency. The letters H, M, and L indicate high, medium- and low-capacity impellers.

The horsepower curve is determined by the efficiency, the head, and the capacity of the pump. It cannot be changed without affecting one of or all the factors. When the efficiency, head, and capacity are known, the horsepower can be computed by the formula:

$$\text{Horsepower} = \frac{\text{gallons per minute} \times \text{head}}{3,960 \times \text{efficiency}}$$

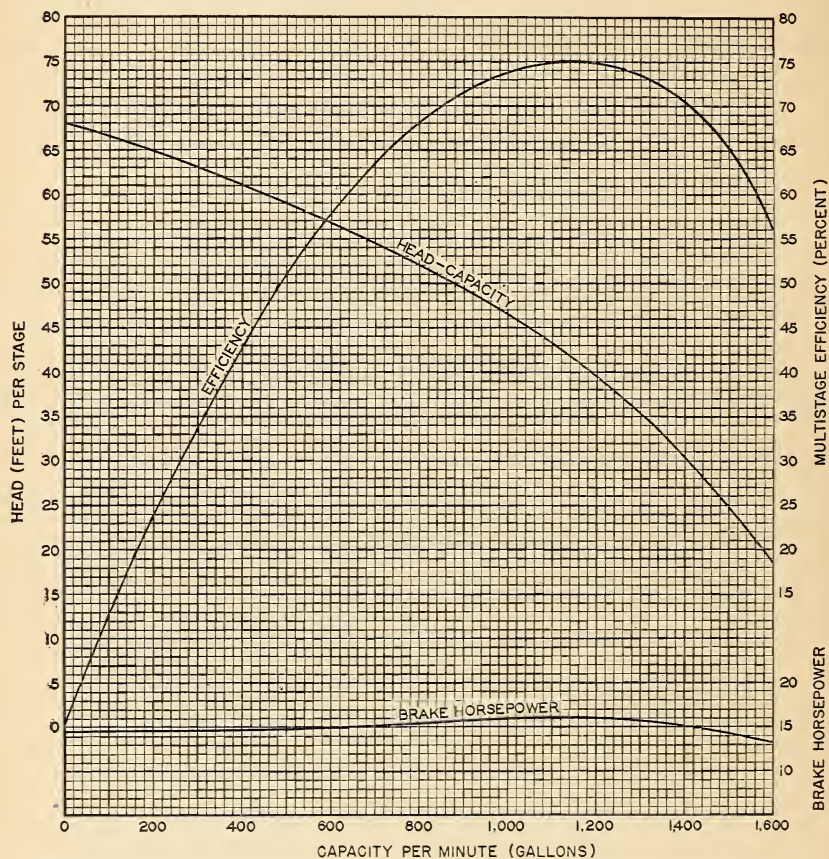


FIGURE 5.—Performance curves of a deep-well turbine of the mixed-flow type, speed 1,750 r. p. m. The horsepower curve is practically constant throughout the range capacity of the pump.

The horsepower can be determined graphically by means of figure 7. It is desirable that the horsepower curve have its peak at the discharge that gives the maximum pump efficiency. In some pumps the horsepower increases as the head decreases and the quantity of water pumped increases, while in others conditions are the reverse. In either case there will be less danger of overloading and thus injuring the power unit if the peak of the horsepower curve is close to the same discharges as those marked by the peak of the efficiency curve and the normal operating condition.

The head against which a centrifugal pump will operate, the discharge, and the power required to drive the pump are functions of the speed and the diameter of the impeller. According to the theory of centrifugal pumps, the discharge is proportional to the speed, the head is proportional to the square of the speed, and the horsepower required is proportional to the cube of the speed. In other words, if the speed is doubled, the discharge will be doubled, the head will be quadrupled, and the horsepower will be multiplied by eight. These relations apply to normal operating conditions and are only approximately correct. Hence, there may be considerable error in the results when the difference between the speeds is large. Under these conditions a new performance curve for different speeds based

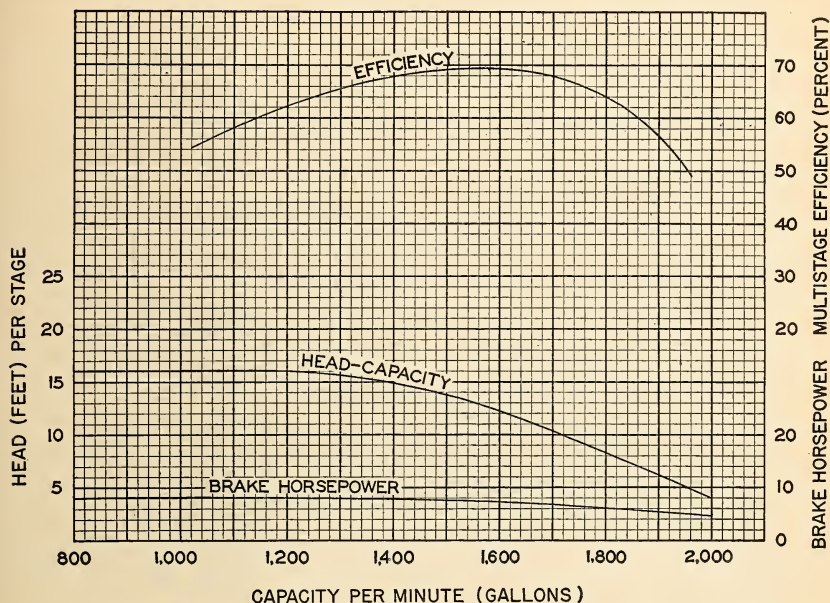


FIGURE 6.—Performance curves of a deep-well turbine of the screw or propeller type, speed 1,750 r. p. m. The horsepower decreases as the head decreases.

on their laboratory-test data should be obtained from the pump manufacturers.

The discharge of a screw or propeller pump is a function of the speed, the diameter of the propeller, and the pitch. Mixed-flow pumps have characteristics similar to those of the centrifugal and the propeller types. However, these two types do not follow the laws of centrifugal pumps. If the characteristics of a mixed-flow pump approach those of a centrifugal type, its performance will be somewhat like that of a centrifugal pump, but if it approaches the propeller type, relations which are known to hold for the centrifugal cannot be applied to the mixed-flow type.

The discharge of a particular pump when operating at a definite speed and against a definite head depends on the efficiency as well as the physical dimensions of the pump. The efficiency of a pump is

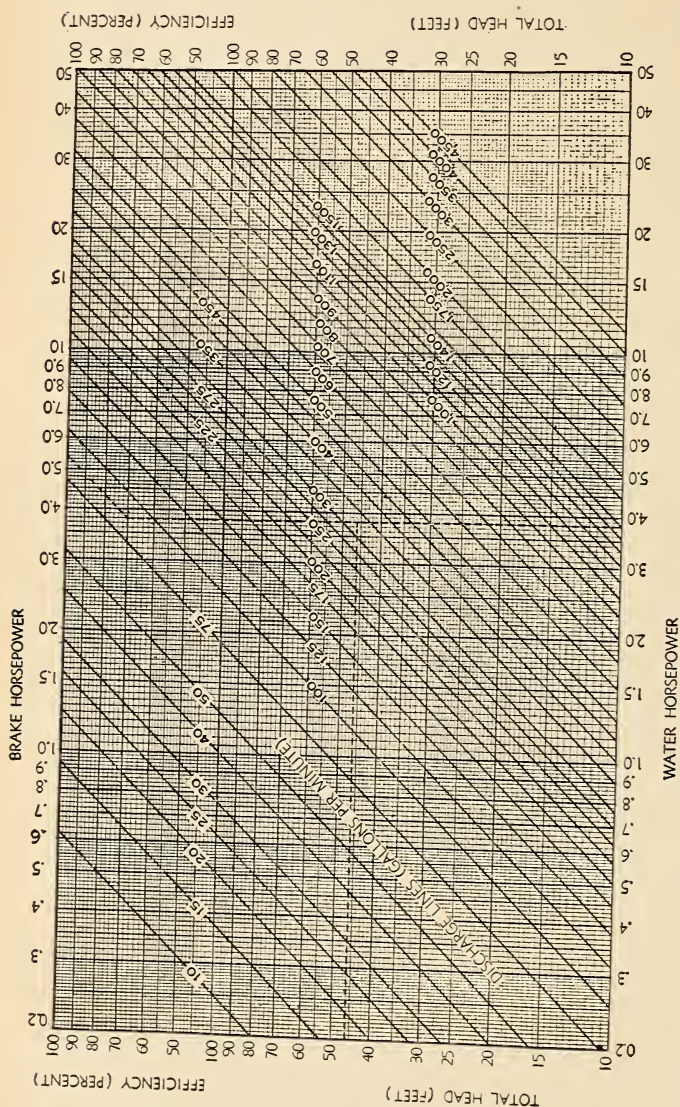


Figure 7.—Horsepower chart. To determine water horsepower enter chart on left side at proper head and proceed horizontally to right to the desired discharge as indicated by the diagonal lines. Directly below this intersection is the water horsepower. To find the brake horsepower required for a given efficiency, proceed vertically until the proper percentage line is reached and then go diagonally to the right parallel to a given efficiency line until the top of the chart is reached. This is the required brake horsepower. The dash lines show the solution of an example when the head is 46 feet, the discharge is 325 g. p. m., and the efficiency of the pump is 70 percent. The solution by the method yields 3.8 for the water horsepower and 5.4 for the brake horsepower. If the brake horsepower, efficiency, and total head are known, it is possible by similar procedure to determine the quantity which can be pumped.

the relation of the water horsepower produced to the horsepower required to drive the pump; that is, the brake horsepower. Thus

$$\text{Efficiency} = \frac{\text{water horsepower}}{\text{brake horsepower}}$$

and the efficiency is usually expressed as a percentage although the ratio is used in all computations.

$$\text{Water horsepower} = \frac{\text{gallons per minute} \times \text{total head}}{3,960}$$

also

$$\text{Water horsepower} = \frac{\text{cubic feet per second} \times \text{total head}}{8.8}$$

The brake horsepower is the power input to the pump shaft. On a direct-connected motor-driven unit it is easily determined from readings on the watt-hour meter if the motor efficiency is known. If the motor is belt-connected, a correction must be made for the belt loss. The brake horsepower of engine-driven units cannot be determined easily. A Prony brake or other type of dynamometer must be used. For this reason accurate information as to the efficiency of engine-driven plants is seldom obtained except in terms of fuel consumed.

The total head referred to in the formulas is the difference in elevation between the water in the well or other source and the center of the discharging stream, plus the pressure head if the stream is discharging under pressure (as where sprinklers are used), plus certain losses. When determining the pump efficiency of a horizontal centrifugal pump or the bowl efficiency of a deep-well turbine, these losses include the velocity head (see p. 63), the loss of head at entrance and in the pipe fittings, and the friction in the suction and discharge pipe and in the discharge pipe fittings beyond the pump. However, when determining the field efficiency of a deep-well turbine the losses included in the total head are the friction in the discharge pipe and in the discharge pipe fittings beyond the pump discharge elbow. Losses in the suction pipe and the pump column are not included, neither is the loss of head at entrance nor the velocity head. In the deep-well turbine these losses are charged to pump efficiency, whereas they are considered as part of the head in a horizontal centrifugal pump.

From the foregoing discussion it is evident that the field efficiency of a deep-well turbine depends in part on the length and diameter of the suction pipe and discharge column. This fact should be given consideration when comparing its efficiency with that of a horizontal centrifugal.

The over-all efficiency of a plant is the ratio of the water horsepower and the power input to the motor or engine. Water horsepower is computed in the manner indicated for determining pump efficiency of a horizontal centrifugal pump and the field efficiency for a deep-well turbine. The power input to the motor is measured by a watt-hour meter. The over-all efficiency will be less than the pump or field efficiency by the amount of the motor losses and power transmission losses.

The over-all efficiency of an engine-driven plant is usually expressed as the pounds, gallons, or cubic feet of fuel used by the engine in pumping 1 acre-foot against a head of 1 foot. If the energy content of the fuel were known, it would be possible to determine the horsepower input to the motor, but this information would not be of particular value to the farmer because he is interested primarily in the amount of fuel it takes to run his engine or the number of kilowatt-hours of electricity to run his motor. This is the basis on which the farmer should purchase his plant. It is the least likely to cause controversies because the fuel or kilowatt-hour consumption, the water pumped, and the head are all factors that can be measured in the field with reasonable accuracy.

METHOD OF CHANGING PUMPS TO FIT NEW CONDITIONS

Although as a rule the pump purchaser is not directly concerned with the theoretical relation of the speed and diameter of impeller to the head, discharge, and power, since these relations are worked out in the engineering department of the pump company, it sometimes happens that conditions change after the pump has been in operation for some time, or the conditions were not correctly determined in the first place. Under these circumstances, if time is not available or if the expense involved is too great, it may not be desirable to return the pump. It is then important to know what to do.

If the capacity of the well is less than the capacity of the pump, it is possible to determine approximately how much the speed of a belt or engine-driven unit should be reduced or how much the impeller of a direct-connected motor-driven unit should be decreased. For example, if an engine-driven deep-well turbine had a capacity of 1,420 g. p. m. when operating at a speed of 1,450 r. p. m. against a head of 52 feet, as shown by the performance curve (fig. 8), and it was installed in a well which after several days of pumping did not yield enough water to supply the pump, it would be necessary to decrease the capacity of the pump by throttling, by reducing the speed, or by turning down the impeller. Of these methods, changing the speed is the most satisfactory in engine-driven plants.

Before attempting to make any changes, the actual capacity of the well when operating against the desired head should be determined. This can be done by partially closing the valve in the discharge line if one is available, or by driving a tapered plug of rectangular section into the outlet pipe until the discharge is decreased the desired amount; that is, until the draw-down is reduced to a safe value that does not make the pumping head exceed the economic limit of lift and that does not cause the pump to lose its prime. When the draw-down becomes constant after pumping has continued several hours at this rate, the discharge should be measured by a Parshall flume (13) or similar means (2).

Suppose the well will deliver 1,040 g. p. m. when the lift, exclusive of the head built up by throttling, is 55 feet. The head that the pump has to operate against to deliver 1,040 g. p. m. at 1,450 r. p. m. can be determined by reference to the performance curve (fig. 8) furnished with the pump. This head is 60 feet. There are several methods by which the correct speed of the pump for this condition can be determined. In this case, the performance curve for the pump

at the 1,750 and 1,150 speeds is available (fig. 8). To find the proper speed, plot the point for the 55-foot head and the 1,040 g. p. m. discharge on the performance chart. This point (fig. 8) falls at the intersection of the vertical lines through 1,040 g. p. m. and the horizontal line through 55 feet. It is about one-fifth the way between the 1,450 r. p. m. line and the 1,150 r. p. m. line. Since the difference between 1,450 and 1,150 is 300, one-fifth the distance is one-fifth of 300, or 60 revolutions, which, when subtracted from 1,450, gives 1,390 r. p. m. as the proper speed.

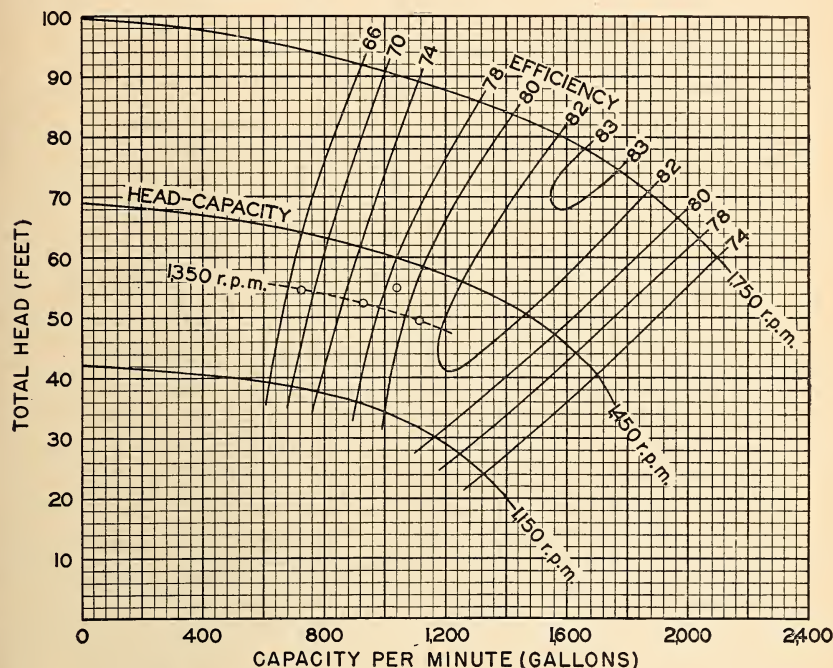


FIGURE 8.—Performance of a centrifugal pump when operating at speeds of 1,150, 1,450 and 1,750 r. p. m. (19). Performance at intermediate speeds can be determined by interpolation as explained on page 17. Dotted line is computed head-capacity curve at speed of 1,350 r. p. m.

Suppose that the head-capacity curve is available for only the 1,450 r. p. m. speed. When this occurs, it is possible to determine the correct speed for the conditions of the example by plotting the head-capacity curve for some speed less than 1,450 r. p. m., utilizing the relations (p. 13) between the head, speed, discharge, and horsepower of centrifugal pumps. For example, take the point identifying discharge 800 g. p. m. and head 63 feet, from the 1,450 r. p. m. curve. At the

1,350 speed the discharge X , since $\frac{X}{800} = \frac{1,350}{1,450}$, will be $\frac{1,350}{1,450} \times 800 = 745$ g. p. m. and the head Y , since $\frac{Y}{63} = \left(\frac{1,350}{1,450}\right)^2$, will be $\left(\frac{1,350}{1,450}\right)^2 \times 63 = 54.6$ feet. Plot this point on figure 8 as shown. Take another point on

the 1,450 r. p. m. curve to the right of the first one chosen, say discharge 1,000 g. p. m. and head 60.5 feet. At the 1,350 speed the discharge will be $\frac{1,350}{1,450} \times 1,000 = 931$ g. p. m. and the head will be $\left(\frac{1,350}{1,450}\right)^2 \times 60.5 = 52.4$ feet. Plot this point on figure 8. Next, take the point for discharge 1,200 g. p. m. and head 57 feet on the 1,450 r. p. m. curve. For the 1,350 speed the discharge will be $\frac{1,350}{1,450} \times 1,200 = 1,117$ g. p. m. and the head will be $\left(\frac{1,350}{1,450}\right)^2 \times 57 = 49.4$ feet. Plot this point and draw a line parallel to the 1,450 r. p. m. curve through the three points. Then plot the point for discharge 1,040 g. p. m. and head 55 feet, which is the condition for which the proper speed is sought. It will be observed that this point falls about midway between the 1,450 and 1,350 r. p. m. curves. Since this interval between the two curves is 100 revolutions per minute, one-half the distance is 50 r. p. m. and the desired speed is 1,450 - 50 or 1,400 r. p. m. By the method first used the speed was 1,390 r. p. m.; the agreement is therefore fairly close.

Sometimes the pump performance curve is not available; in that case it is necessary to measure both the head and the discharge when making the test to determine what the well can safely deliver. By tapping into the discharge pipe between the valve and the discharge elbow and attaching a glass gage by means of a rubber tube, it is possible to measure the head above the point of connection. The vertical distance between the water surface in the well and in the glass tube is the total head to be considered. The discharge is measured as before. By test it is found that the pump has to be throttled by closing the discharge valve until the pressure read on the gage glass attached to the discharge pipe is 5 feet and the distance to water 55 feet when the water pumped is all the well will safely deliver without exceeding the economic limit of lift. Suppose the discharge under these conditions is 1,040 g. p. m. as before. Since the head is proportional to the square of the impeller speed, it is possible to determine the speed at which the pump must run to deliver approximately 1,040 g. p. m. against a head of 55 feet (which according to the test was found to be the safe limit) by setting up the following equation:

$$\frac{X^2}{1,450^2} = \frac{55}{60}, \text{ in which } X \text{ is the speed. Then } X = 1,450 \sqrt{\frac{55}{60}} = 1,390 \text{ r. p. m.}$$

At this speed, the discharge will be less than 1,040 g. p. m. because the discharge is proportional to the speed. Actually, it will be $\frac{1,390}{1,450} \times 1,040$, or 1,000 g. p. m. In this case the discharge differs from the results obtained by the first solution of the problem. The first method is probably the more accurate because it is based on values determined experimentally.

To find the proper diameter of pump pulley for this speed, assuming that 1,390 is the correct value, multiply the diameter of old pulley by $\frac{1,450}{1,390}$. If it is the plan to change the engine pulley diameter,

it should be reduced by the ratio of $\frac{1,390}{1,450}$. The diameter of the pump

pulley has to be increased or the engine pulley reduced to decrease the speed of the pump.

If the capacity of the well is greater than the discharge of the pump, the amount that the speed must be increased to give the desired discharge can be determined by similar computations.

Direct-connected motor-driven units can best be made to fit new conditions by changing the diameter of the impeller, because the speed is fixed. Take, for example, a direct-connected motor-driven unit with 9-inch impeller designed to deliver 1,300 g. p. m. at a speed of 1,750 r. p. m. against a head of 55 feet, as shown in figure 9. Suppose as before, it pumps the well dry, and that by test the pump will

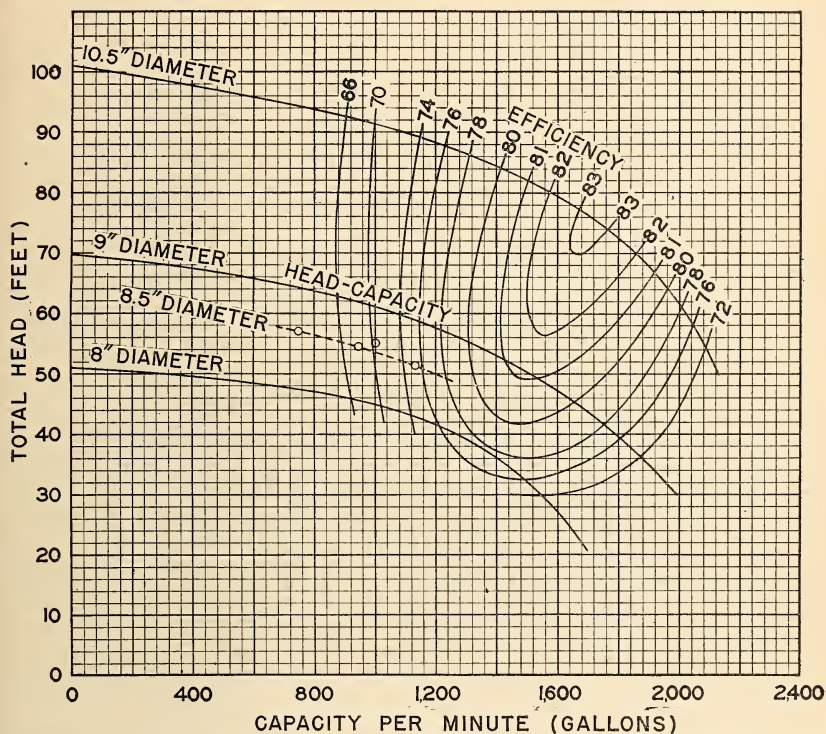


FIGURE 9.—Performance of a centrifugal pump with propellers of different diameters when operating at a constant speed of 1,750 r. p. m. (19). Performance at other diameters can be determined by interpolation as explained on page 19. Dotted line is computed head-capacity curve for 8.5 inch impeller.

safely deliver 1,000 g. p. m. without exceeding the economic limit of lift. Suppose also for this discharge the head exclusive of that built up by throttling the pump to reduce the discharge is 55 feet and the head due to throttling as measured by a glass gage on the discharge pipe is 6 feet. When there is available a diagram such as figure 9, which shows the head-capacity for impellers of different diameters, it is possible to determine what the diameter should be for delivery of 1,000 g. p. m. against a head of 55 feet, by plotting this point on the diagram and noting where it falls with reference to the head-capacity curves. The point is approximately one-third the distance

from the 9-inch impeller curve to the 8-inch impeller curve. Since the distance between the curves is equivalent to 1 inch in diameter, one-third the distance is 0.33 inch and the required diameter of impeller is $9.00 - 0.33$, or 8.67 inches.

Usually the performance chart of a pump shows only the head-capacity curve for an impeller of the diameter used in the pump. Where this occurs the proper diameter of impeller to meet the new conditions can be determined by plotting the new head capacity for an impeller of slightly less diameter, say 8.5 inches, by following the procedure explained on page 17.

Since the head against which the pump will operate varies as the square of the speed and since the peripheral speed of the impeller is directly proportional to the diameter, the head also will vary as the square of the diameter. The discharge is directly proportional to the speed and therefore to the diameter for the same reason.

To locate the head-capacity curve for an 8.5-inch impeller, choose three points on the head-capacity curve for the 9-inch impeller on that portion in which the problem falls. Take the discharges 800, 1,000 and 1,200 g. p. m. for which the corresponding heads are 64, 61, and 57.5 feet (fig. 9). To find the discharge for the first point when

impeller is 8.5 inches in diameter, set up the equation $\frac{X}{800} = \frac{8.5}{9.0}$

in which X is the discharge. Then $X = 756$ g. p. m. The corresponding head is obtained by solving for Y in equation $\frac{Y}{64} = \frac{8.5^2}{9.0^2}$, from

which Y is found to be 57.2 feet. By similar computations the discharge and head for the second point are 945 g. p. m. and 54.4 feet; and for the third point, 1,134 g. p. m. and 51.3 feet. These new points are then plotted on the performance chart (fig. 9) and a line is drawn through them parallel to the head-capacity curve for the 9-inch impeller. It will be noted that the point for discharge 1,000 g. p. m. and head 55 feet, for which the impeller diameter is sought, is about one-quarter of the interval between the two curves away from the curve for the 8.5-inch impeller. Since this interval is 0.5 inch, the diameter of the new impeller is one-quarter of 0.5 inch or 0.12 more than 8.5; that is, 8.62 inches. By the method previously described the diameter was found to be 8.67 inches. This diameter is probably more nearly the true value because it is based on the performance of the different impellers as shown by pump tests, whereas the diameter found by the second method is based partly on theoretical relations known to be only approximately correct.

If no performance curve of the pump is at hand the diameter of the impeller for the condition of the problem can be determined approximately from the known facts and the theoretical relations of head, discharge, and diameter. Suppose that the impeller diameter was 9 inches as before, and that it was found by test that the well would safely deliver 1,000 g. p. m. when the pump was throttled until the head was 61 feet and the head excluding that built up by throttling was 55 feet, this being the head against which the cut-down propeller must operate when delivering 1,000 g. p. m. Since the head is proportional to the square of the diameter of the impeller, $\frac{55}{61} = \frac{X^2}{9^2}$,

It is readily determined that $X=8.55$ inches. For this diameter, the discharge is found by setting up the equation $\frac{Y}{1,000}=\frac{8.55}{9.00}$, in which Y is the discharge. Solving, $Y=950$ g. p. m. Decreasing the diameter of the impeller causes changes in the water passages in the impeller which also affect the discharge; consequently the diameter probably will not have to be decreased quite so much as shown. To be on the safe side since the impeller cannot be built up if too much is cut off, the diameter might be cut to, say, 8.7 inches; then if the pump still delivers too much, 8.6 inches might be tried.

The changes in impeller diameter or speed necessary to make the pump fit the well also change efficiency and horsepower. Since the pump chosen is usually designed to operate at maximum efficiency for the discharge and head existing at the well at the time the pump is purchased, changes in impeller speed or diameter usually result in lower pump efficiency. If the changes required are small, the reduction in efficiency will probably be small also. The horsepower required will depend on whether the capacity is increased or decreased, because these changes will probably be greater than the changes in efficiency.

Since the horsepower varies as the cube of the speed and the peripheral speed of the impeller is directly proportional to the diameter, the horsepower varies also as the cube of the diameter. In the example just considered the pump will deliver 1,000 g. p. m. against a head of 61 feet when the discharge is throttled by the gate valve; therefore, the water horsepower developed is $\frac{1,000 \times 61}{3,960} = 15.4$. (See p. 15.) The efficiency for this condition is not known, but since the efficiency curve is flat in the region in which it is designed to operate most economically, it may be assumed that the efficiency will be only slightly less than when 1,300 g. p. m. were being pumped. This information is supplied by the manufacturer with the pump. Assuming that it is, say 80 percent for this delivery and slightly less or 75 percent when delivering 1,000 gallons against a head of 61 feet, the horsepower required is then equal to $\frac{15.4}{0.75}$ or 20.5. When the diameter of the impeller is reduced from 9 inches to 8.55 inches it will deliver 950 gallons against a head of 55 feet. The horsepower required will be $\frac{8.55^3}{9.00^3} \times 20.5$, or 17.6.

Since the head and discharge are known and the efficiency can be approximated it is also possible to compute the horsepower by substituting these data in the formula on page 12. The discharge, as previously determined, is 950 g. p. m. when the impeller diameter is 8.55 inches. If it is assumed that the efficiency is the same as before, then the horsepower required, computed by the formula, is $\text{Horsepower} = \frac{950 \times 55}{3,960 \times 0.75} = 17.6$, which checks the result obtained by the other method.

When it is necessary to make changes in pumps of the mixed-flow or axial-flow type, the relations are so involved because of the combination of centrifugal and screw action, that it is not possible to de-

termine how much the speed or diameter should be changed to fit the new conditions except by reference to diagrams, such as figures 8 and 9. These diagrams, however, are for definite pumps and cannot be used to solve problems concerning other pumps. It is for this reason that all data and charts supplied by the pump manufacturers should be carefully kept for future reference. If a pump-performance curve is not furnished with the pump, a copy should be demanded. This information will be useful if there is ever any trouble with the pump.

MOTIVE POWER

The choice of power unit for a small pumping plant is usually restricted to electric motors and internal-combustion engines. Small steam plants are not so economical as electric motors or internal-combustion engines. They require much more attendance and consequently are seldom used to operate small pumping plants. Current

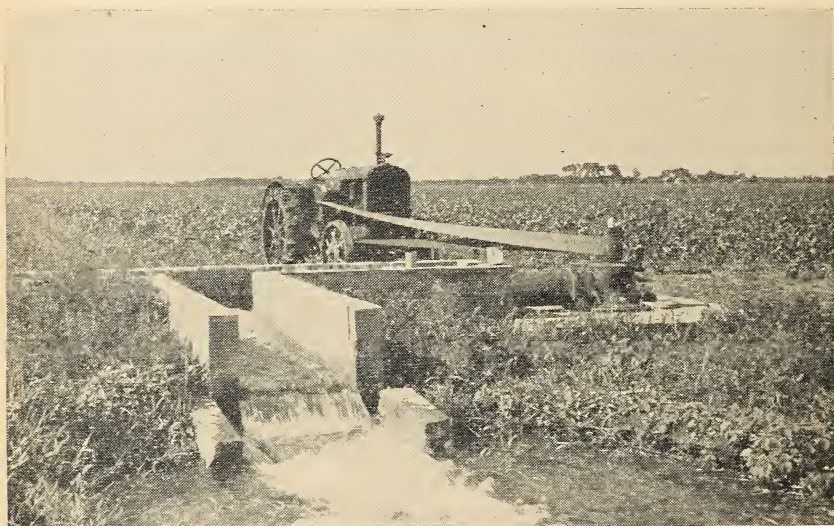


FIGURE 10.—Heavy tractor driving turbine pump by means of quarter-turn flat belt. Note simplicity and effectiveness of installation.

wheels have been utilized to some extent, but the high cost of maintaining them, and their small capacity, have usually led to their replacement by more satisfactory equipment. The use of large windmills has been abandoned for similar reasons. Windmills, however, still fill a need in providing power for pumping water for gardens from small wells having capacity less than 50 g. p. m.

Where electric power is available and the rates are reasonable, the electric motor is the most dependable and probably the cheapest source of power for a pumping plant. Its long life, low first cost and repair charges, freedom from starting troubles and necessity for attendance, quietness of operation, and low fire hazard will, in most cases, balance the savings in cost of operation obtainable with Diesel, natural gas, and special fuel engines. Another important advantage of electric motors is that they will operate at practically their original efficiency through-

out their entire lives if they are given reasonable care. Where, because of lack of electric power or its high cost, it is necessary to use some type of internal-combustion engine the choice of type will be governed by the first cost, operating conditions, and the cost of the different kinds of fuel required. If the farmer has a tractor with a power-take-off which is not used during the pumping season, it should be considered a source of power for the plant. Its use will materially reduce the investment in the plant during the development period and may be a deciding factor in its success. Later, when the project is fully developed and requires a long period of operation each season, it may be desirable to substitute a power unit primarily for running the pump. Figure 10 shows a successful tractor-driven plant.

ELECTRIC MOTORS

The most common type of motor used on pumping plants is the three-phase alternating-current induction motor. Motors of this type do not require brushes. Single-phase motors are used for small loads of 5 horsepower or less. These small motors have special starting equipment requiring a commutator and brushes. Standard motor sizes are 1, 1½, 2, 3, 5, 7½, 10, 15, and 20 hp. The approximate cost of motors of this type is given in table 4.

Induction motor speeds are determined by the number of poles of the motor and the number of cycles of the current by the formula :

Revolutions per minute = $\frac{\text{number of cycles} \times 120}{\text{number of poles}}$

Since most power plants now produce only 60-cycle current, direct-connected pumps are designed to operate on 60-cycle speeds which are 3,475, 1,760, 1,160, and 870 r. p. m. The corresponding 50-cycle speeds are 2,900, 1,465, 965, and 725 r. p. m. The differences between these speeds and those computed by the formula are caused by the slight lag that occurs when the motor is loaded.

TABLE 4.—Delivered prices of 3-phase 60-cycle, 220-440 volt, squirrel-cage, induction motors complete with base, pulley, and line starter or compensator

| Horsepower | Speed in revolutions per minute | | | | | | | | |
|-----------------|---------------------------------|---------------------------|--------------------|--------|--------------------------|--------------------|--------|--------------------------|--------------------|
| | 1,760 | | | 1,160 | | | 870 | | |
| | Weight | Pulley size. ¹ | Price ² | Weight | Pulley size ¹ | Price ² | Weight | Pulley size ¹ | Price ² |
| | Pounds | Inches | Dollars | Pounds | Inches | Dollars | Pounds | Inches | Dollars |
| 2 ³ | 190 | 4×3 | 72 | 270 | 4½×4 | 128 | | | |
| 3 ³ | 190 | 4×3 | 90 | 285 | 4½×4 | 153 | | | |
| 5 ³ | 270 | 4½×4 | 128 | 455 | 5×4 | 184 | | | |
| 7½ | 320 | 5×4 | 134 | 425 | 6×5 | 161 | 505 | 8×6 | 191 |
| 10 | 425 | 6×5 | 161 | 505 | 8×6 | 184 | 670 | 9×7 | 226 |
| 15 | 485 | 8×6 | 184 | 505 | 9×7 | 226 | 825 | 10×7 | 268 |
| 20 | 655 | 9×7 | 230 | 840 | 10×7 | 286 | 925 | 10×7 | 326 |
| 25 | 675 | 9×7 | 255 | 925 | 10×7 | 326 | 1,150 | 11×9 | 368 |
| 30 | 925 | 10×7 | 326 | 1,150 | 11×9 | 368 | 1,165 | 11×9 | 442 |
| 40 ⁴ | 1,520 | 11×9 | 516 | 1,635 | 11×9 | 575 | 1,840 | 12×10 | 633 |

¹ First figure is pulley diameter, second figure is belt width.
² Prices are for year 1940.
³ Single phase, 110-220 volts.
⁴ Equipped with compensator, others with line starters.

The latest-type induction motors under 40 hp. in size are specially wound so that they may be equipped with line starters to eliminate the necessity of having compensators. Induction motors are usually built to operate on 120, 220, or 440 volts, the 220- and 440-volt motors being most common. The 220-440 volt motors are designed to operate on either voltage by simply changing the connections. The size of the motor per horsepower decreases as the voltage increases, but voltages higher than 440 are not recommended for pumping plants because of the danger of accidental electrocution.

The size decreases also as the speed increases. However, since the 3,475 speed is too high for most pumps, the 1,760-speed motor is used most frequently. Because of its more general use, it is usually the cheapest. The efficiency of three-phase induction motors when fully loaded is between 80 and 90 percent. Single-phase motors have a lower efficiency than three-phase motors and small motors have a lower efficiency than large motors. There is also a decrease in efficiency if motors are not fully loaded. Well-made motors will operate satisfactorily under a continuous overload of 10 percent if properly ventilated. It is usually more economical to overload a motor slightly than to install the next larger size, on account of the lower electric rate (16) as well as the saving in first cost of the motor. If there is any doubt as to the size of motor required, the power company or the motor manufacturer should be consulted.

Induction motors have a high starting torque when started on full line voltage. For this reason the motor can be direct-connected to the pump without the use of a clutch. This is not true of synchronous motors, but since they are not used on small plants the method of starting them need not be discussed. Induction motors will run in either direction, and the direction of operation of three-phase motors may be changed at any time by merely interchanging two of the leads. The direction of rotation of a single-phase motor may be reversed by interchanging the leads in the starting mechanism.

As previously stated, the speed of an induction motor is fixed by the number of poles and the number of cycles. In all direct-connected units it is therefore important that the pump be designed to operate at the motor speed. If conditions change, the pump will have to be changed to meet the new conditions. When the water level fluctuates through wide limits as in some streams and reservoirs, it may prove economical to use a variable-speed motor. These motors are more expensive than constant-speed motors and have to be adjusted to give the proper speed for the different water levels. Their use will probably prove most satisfactory on the larger installations where there is likely to be someone in attendance who knows how to adjust the speed to fit the head against which the pump is operating.

Induction motors may be equipped with either sleeve or ball bearings, the latter type being slightly more efficient as well as more expensive. Vertical motors always have ball bearings because they are easier to lubricate than sleeve bearings in the vertical position and are more satisfactory for carrying vertical thrust. Vertical motors for direct-connected deep-well turbines and screw pumps usually have hollow shafts through which the pump shafts extend. This arrangement permits putting the adjusting unit and the locking mechanism on top of the motor where it is easily accessible. The

locking mechanism is made so that it will transmit power from the motor to the pump in only one direction in order to avoid the possibility of unscrewing the pump shaft if for any reason the motor direction is reversed. Horizontal motors with grease-packed ball bearings may be run in the vertical position. By mounting the motor in this position it is possible to avoid the use of quarter-turn belts on vertical pumps where it is necessary to use belts to obtain the proper speed.

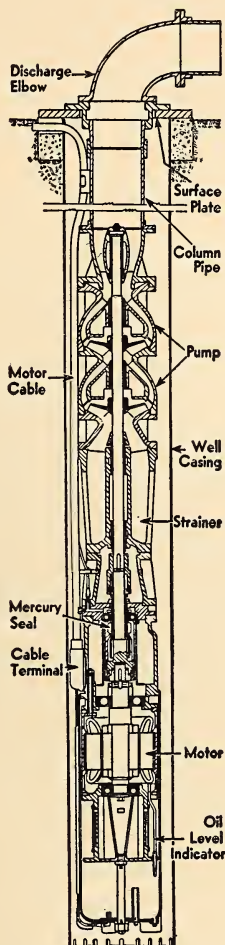


FIGURE 11.—Section of submersible motor installation in deep well.

Most vertical motors are now made weatherproof; nevertheless, although they will operate satisfactorily without cover, shelters should be provided to protect such motors from vandals. A new type of vertical motor (fig. 11) has a watertight-metal case; thus it may be direct-connected to the pump bowls beneath the water surface in deep wells thereby eliminating the long shaft required in surface installations. The submersible motors are made smaller in diameter and much longer than ordinary motors so that they may be inserted in wells of the usual

diameter. These motors have been in use about 10 years and seem to be giving satisfactory service. Their use should be given consideration in very deep or crooked wells. Plants with submersible motors compare favorably in price and efficiency with those having the conventional types of motors.

Motors should be equipped with low voltage release and thermal overload relay. The low voltage release shuts off the motor in case the power is cut off or the voltage drops enough to endanger the motor. Where many motors are operated with automatic switches that start them when the power comes on again, some of the switches are of the delayed operation type and thus prevent all the motors coming on at once with a resulting serious overloading of the system. The thermal overload relay shuts off the motor when it is drawing too much current due to overload or other causes.

CHOICE OF ELECTRIC MOTOR TO FIT CONDITIONS

The choice of motor to fit given conditions depends on the electric rate schedule as well as the lift, capacity of the well, and area to be irrigated. For example, suppose that a farmer has an 80-acre tract for which the irrigation requirement is 2 acre-feet per acre or a total of 160 acre-feet during the period May 15 to September 15. Suppose also that he has a well that will yield 900 g. p. m. with a draw-down of 20 feet and 1,350 g. p. m. with a draw-down of 30 feet. For these conditions assume that the total heads are respectively 40 feet and 50 feet.

If the farmer chooses a pump with a capacity of 900 g. p. m. and operates it 24 hours a day he will pump 4 acre-feet each day (450 g. p. m. for 24 hours equals approximately 2 acre-feet); whereas, if he pumps 12 hours a day he will pump only 2 acre-feet each day. Since 160 acre-feet is required, it will take 80 days of 12 hours each to pump this quantity of water. Owing to variation of the demand for water throughout the season, the pump will have to be operated only part of the time at the beginning and end of the irrigation season, and during the period of maximum demand it may be necessary to run the plant every day and even 24 hours a day at times to supply all the water required.

In order to irrigate the ground rapidly and eliminate the necessity of irrigating at night, most farmers like to work with as large a stream as they can manage. If the pump chosen has 1,350 g. p. m. capacity and is operated 24 hours a day, it will deliver 6 acre-feet each day; if operated 12 hours a day, it will deliver 3 acre-feet daily. At the 12-hour rate it will take the pump $53\frac{1}{3}$ days to deliver 160 acre-feet. It is apparent that to irrigate his farm with the larger pump the farmer will have to operate his plant much less than if he uses a 900 g. p. m. pump, and the labor required will be correspondingly lessened.

However, there are disadvantages offsetting the advantages of using a large plant. The larger pump draws down the well an additional 10 feet, which increases the lift 25 percent. This increases the cost of pumping an acre-foot by approximately the same proportion. The cost of the plant required to pump 1,350 g. p. m. is considerably higher than the cost of one to pump 900 g. p. m. Moreover, a larger motor will be required and therefore the cost of electricity per kilowatt-hour will usually be higher because of the demand charge included in most

electric rates. The demand charge may be incorporated in the rate as a flat charge per horsepower per season, as a guaranteed minimum payment for power, as a higher charge of the first power used, or as a combination of these charges.

As an indication of the way the electric rate affects the cost of power when two different sizes of motors are considered, take the examples just discussed and assume that the over-all efficiency of the plants in both cases will be 60 percent and the motor efficiencies 90 percent. The horsepower required for the small plant will therefore be

$$\frac{900 \times 40}{3,960 \times 0.60} = 15.1 \text{ hp.}$$

and the motor output will be 13.59 hp. (See p. 12). For the larger plant the power will be

$$\frac{1,350 \times 50}{3,960 \times 0.60} = 28.4 \text{ hp.}$$

and the motor output will be 25.56 hp.

Although the horsepower required for the small plant is slightly under 15, a 15-hp. motor will be required. For the larger plant, however, a 25-hp. motor will be necessary because the overload on a 20-hp. motor would be too great for satisfactory operation. (See p. 24.) Since 746 watts equal 1 hp. and 1,000 watts equal 1 kilowatt, the small plant will require 11.3 kw.-hr. for each hour of operation; the larger plant will need 21.2 kilowatts. The small plant, as previously shown, will have to operate 80 days of 12 hours each, or 960 hours to deliver 160 acre-feet, and the larger plant 53½ days of 12 hours each or 640 hours to deliver the same quantity. The total kilowatt-hours required will therefore be 960 times 11.3 or 10,848 kw.-hr. for the small plant and 640 times 21.2 or 13,568 kw.-hr. for the larger plant.

Suppose the following electric rate schedule for irrigation pumping is in force in the farmer's territory:

| | <i>Cents per kilowatt-hour</i> |
|---|------------------------------------|
| First 100 kw.-hr. per horsepower connected load per season..... | 5 |
| Next 100 kw.-hr. per horsepower connected load per season..... | 3 |
| Next 100 kw.-hr. per horsepower connected load per season..... | 2 |
| Next 200 kw.-hr. per horsepower connected load per season..... | 1½ |
| All additional kw.-hr. per horsepower connected load per season..... | 1 |
| Minimum charge \$7.50 per horsepower per season. (This is a Colorado rate schedule. Lower rates are in force in California and in some other States.) | |

The annual power charge for the plants according to the rate schedule is as follows:

900-gallon-per-minute plant:

| | |
|------------------------------|----------|
| 1,500 kw.-hr. at \$0.05..... | \$75. 00 |
| 1,500 kw.-hr. at .03..... | 45. 00 |
| 1,500 kw.-hr. at .02..... | 30. 00 |
| 3,000 kw.-hr. at .015..... | 45. 00 |
| 3,348 kw.-hr. at .01..... | 33. 48 |
| | <hr/> |
| | 228. 48 |

1,350-gallon-per-minute plant:

| | |
|------------------------------|---------|
| 2,500 kw.-hr. at \$0.05..... | 125. 00 |
| 2,500 kw.-hr. at .03..... | 75. 00 |
| 2,500 kw.-hr. at .02..... | 50. 00 |
| 5,000 kw.-hr. at .015..... | 75. 00 |
| 1,068 kw.-hr. at .01..... | 10. 68 |
| | <hr/> |
| | 335. 68 |

The saving in the cost of power effected by installing a small plant is apparent from these figures. In addition, since the first cost of the large plant will be considerably greater than that of the small plant, it is clear that the convenience and saving of time required for irrigating with a larger plant would not justify the expenditure required to obtain these advantages. However, there is a limit beyond which it is not feasible to reduce the size of the plant because the discharge may be too small to irrigate with efficiency or to supply the water required by the crops when the maximum demand occurs.

INTERNAL-COMBUSTION ENGINES

Internal-combustion engines are of three types, but only two are at present used extensively in irrigation pumping plants. These are electric-ignition engines that burn gasoline, kerosene, tractor fuel, or natural and artificial gas, and Diesel engines that burn fuel oil. The other type, the hot-spot or semi-Diesel engine, was once in common use in irrigation pumping plants but is no longer of importance.

Modern electric-ignition engines are equipped with spark plugs that fire the fuel charge in the cylinders. Whether these engines operate on gasoline, kerosene, fuel oil, or gas they are essentially the same in design. They differ chiefly in the carbureting equipment because of the wide variation in the volatility of these fuels. Hot-spot or semi-Diesel engines have an element in the head which has to be preheated to start them. After the engine starts to run, the heat of combustion is sufficient to maintain the temperature of the element above the ignition point of the fuel. This type of engine is rapidly being displaced by the new high-speed Diesels, which are lighter in weight and easier to operate.

Diesel engines have no electric-ignition system. The pressure produced in the cylinder by the piston raises the temperature in the combustion chamber to the ignition point of the fuel. A high-compression ratio is necessary to develop sufficient temperature to ignite the fuel. The compression ratio is the relation between the final volume of the charge in the cylinder and the initial volume. For gasoline engines the ratio is about 1 to 5, for gas engines it may be as high as 1 to 8, and for Diesel engines it is about 1 to 16. The high compression ratio in Diesel engines results in a very high pressure in the cylinder at the end of the compression stroke, and for this reason Diesel engines must be made stronger and heavier than other types. High-compression ratios increase engine efficiency and power in all types of internal-combustion engines, and for this reason there has been a definite increase in the compression ratio in gasoline engines in recent years. High compression, however, increases the tendency to knock, and the increase in pressure has been made possible only by the improvement in the antiknock properties of the fuels.

Most internal-combustion engines are of the four-cycle type; that is, every fourth stroke of the piston is a power stroke. However, there are some modern high-speed Diesel engines of the two-cycle type. They are lighter in weight, but they must have an auxiliary source of supply of air under pressure to drive out the burned gases at the end of the power stroke and to provide a new supply of air for the next explosion. In the four-cycle engine the scavenging is accomplished by the

return stroke of the piston. In two-cycle engines an auxiliary cylinder or the crank case is used to develop sufficient air pressure to drive out the burned gases.

Various types of fuel are available for use in internal-combustion engines, and engines are being manufactured which are suitable for each type of fuel. Some may easily be adapted to operate on several types of fuel. The principal fuels now in use are gasoline, kerosene, tractor fuel, distillate, natural gas, and liquid gas. The prices of these fuels vary considerably in different parts of the United States, depending on the location of the source of supply with reference to the pumping plant and to other causes such as freight rates and taxes. The kind of fuel most economical to use depends on its cost and the power produced per unit of fuel, and on the cost of the type of engine required. Distillate is low in price and the fuel consumption of Diesel engines is low, but the cost of a Diesel engine is more than twice that of a gasoline or natural gas engine of equal power. In order to determine whether a Diesel engine will be more economical than other types, it will be necessary to consider the additional investment in the plant. How this is done is explained on page 46.

The amount of fuel consumed per horsepower-hour by an engine in good condition and proper adjustment depends on the kind of fuel, the altitude, the temperature, the speed, and on whether or not the engine is fully loaded. The consumption of different kinds of fuel per brake horsepower-hour at sea level when the air temperature is 60° F. and when the engine is completely equipped and fully loaded is set out in table 5. The fuels listed in table 5 are mixtures of hydrocarbons in some cases of widely varying composition. Consequently, the values given as to their properties may vary considerably. This fact should be taken into account when the table is used. The values given for fuel consumption are based on tests of new engines under laboratory conditions. For field installations they should probably be increased by at least 10 percent.

TABLE 5.—Consumption of different kinds of fuel per brake horsepower-hour at sea level when the temperature of the air is 60° F. and engine is completely equipped and fully loaded, and other pertinent data

| Fuel | Weight per gallon | Den- sity ¹ | Heating value ² | | Fuel required per brake horsepower-hour | | |
|------------------------------|-------------------------|---------------------------|----------------------------|-------------------|--|------------------|-------------------|
| | | | per pound | per cubic foot | | | |
| | <i>Pounds</i> | <i>° B.</i> | <i>B.t.u.</i> | <i>B.t.u.</i> | <i>Pound</i> | <i>Gallon</i> | <i>Cubic feet</i> |
| Gasoline..... | 6.0-6.3 | 59 | 20,750± | ----- | 0.60-0.64 | 0.10 | ----- |
| Kerosene..... | 6.8 | 41 | 19,800 | ----- | .72- .75 | .09 | ----- |
| Distillate tractor fuel..... | 7.0 | 37 | 19,700 | ----- | .72- .75 | .095 | ----- |
| Fuel oil..... | 7.0-7.6 | 24-27 | 19,300± | ----- | ³ .48- .50 | ³ .07 | ----- |
| Butane (liquid)..... | 4.7-4.8 | 115 | 21,000± | ----- | .41- .51 | .10 | ----- |
| Butane (gas)..... | ----- | ----- | ----- | 3,200± | ----- | ----- | 3.0 |
| Natural gas..... | ----- | ----- | ----- | 950-1,150 | ----- | ----- | 12.5 |
| Manufactured gas..... | ----- | ----- | ----- | 500- 800 | ----- | ----- | ⁴ 15.0 |

¹ See p. 35 for explanation of Baumé scale.

² See p. 35 for definition of B.t.u. (British thermal unit).

³ Diesel engine.

⁴ 800 B.t.u. gas.

The power developed by an electric-ignition internal-combustion engine decreases about 3 percent for each 1,000 feet rise in elevation above sea level and about 1 percent for each 10° F. rise in tempera-

ture above 60° . By installing high-compression pistons or cylinder heads, a portion of the loss due to altitude can be regained. If air temperatures over 110° are encountered, provision should be made for cooling the intake manifold. Diesel engines also develop less power in rarefied air, but the amount depends on the design of the engine as it normally draws in an excess of air on the intake stroke. Fuel consumption per brake horsepower increases slightly as the

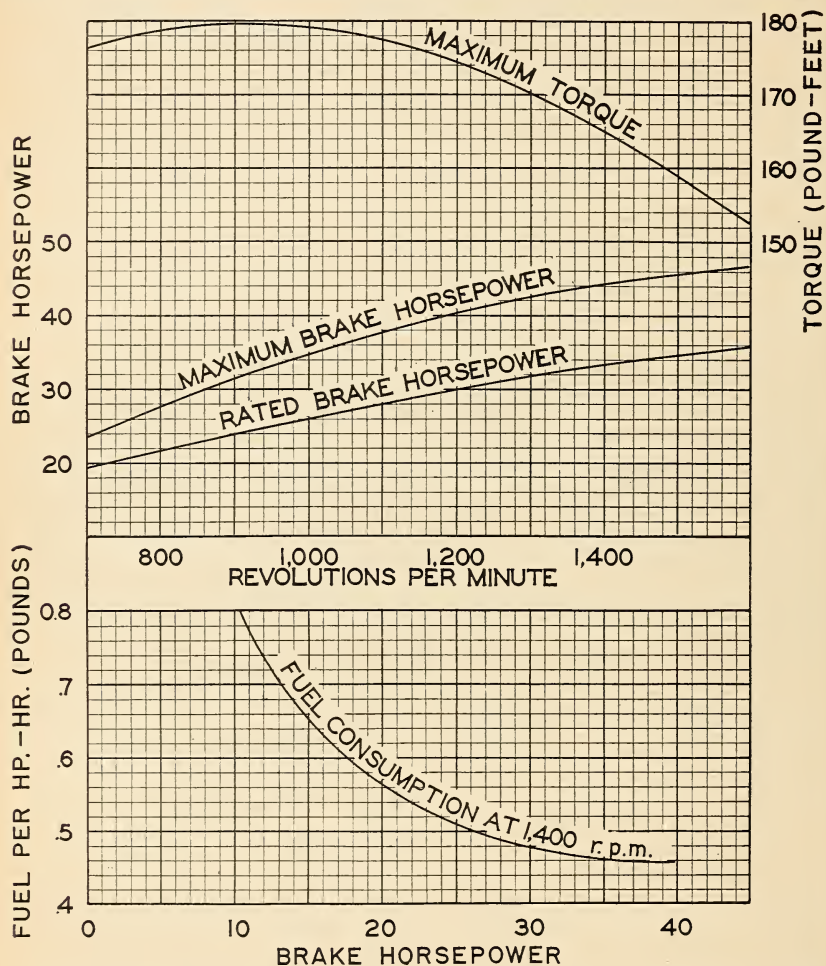


FIGURE 12.—Performance diagram of four-cylinder modern high-speed Diesel engine.

engine speed increases above a certain point, but the power increases almost in direct proportion to the speed. (See figs. 12 and 13.) For this reason it is generally desirable, in order to get the additional power, to disregard the slight increase in fuel consumption. Fuel consumption per brake horsepower-hour increases if the engine is not fully loaded, and as it is not customary to load the engine above 80 percent of its maximum horsepower for continuous operation,

the fuel consumption is usually slightly greater than the values shown in table 5. If the engine is loaded to only a small proportion of the rated horsepower for the speed, the fuel consumption will be considerably increased. Under these conditions it will be more economical to reduce the engine speed until the proper horsepower is developed. By changing the pulleys the proper pump speed can be obtained.

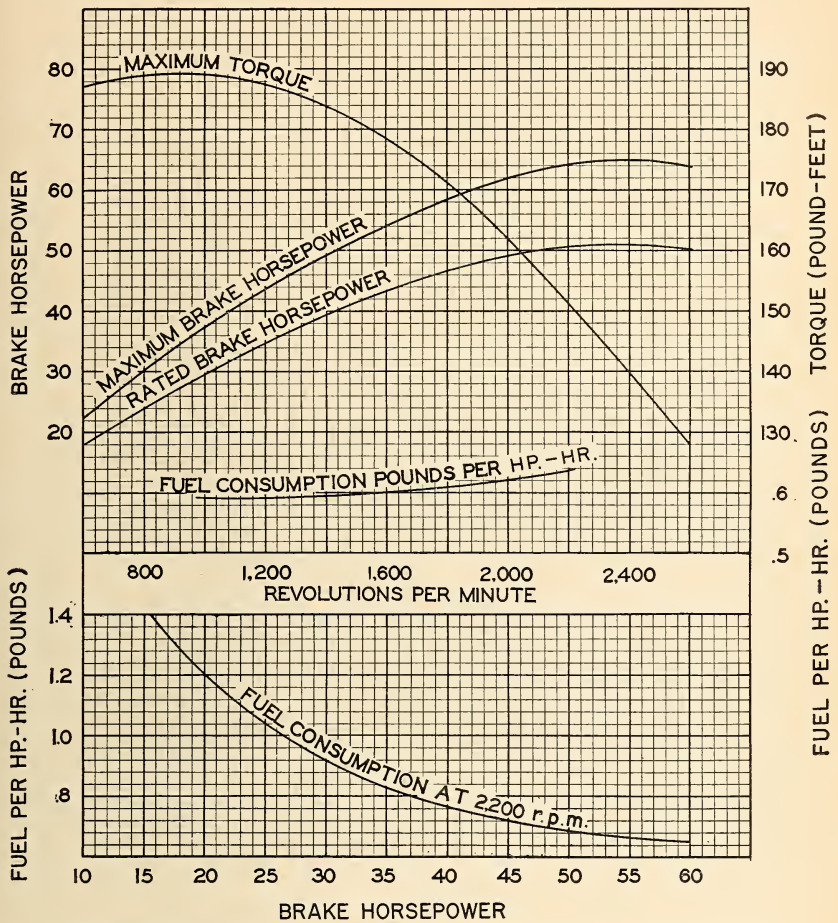


FIGURE 13.—Performance diagram of six-cylinder gasoline-power unit.

To compare the fuel cost per brake horsepower-hour for different kinds of fuel, the fuel consumption shown in table 5 for the specified fuels should be multiplied by the cost of the fuel. Whichever gives the lowest value will be the most economical for use from the standpoint of consumption. Other factors, however, such as first cost of equipment, depreciation, or ease of handling, also influence the cost of operation. Before final decision is reached as to the type of en-

gine and the kind of fuel to be used, all these items should be investigated.

The high-speed lightweight Diesel engines now being manufactured for pumping service and similar uses range from 5 to 120 hp. In speed they range from 800 to 1,500 r. p. m. depending on the make. The engines have from 1 to 8 cylinders; as more power is needed more cylinders are added. Three-cylinder engines are common. The approximate cost of Diesel engines is given in table 6.

TABLE 6.—*Delivered prices of high-speed Diesel industrial power units*

| Unit | Speed | Rating of motor | | Cylinders | Weight | Price ¹ |
|------|-----------------|-------------------|-------------------|---------------|---------------|--------------------|
| | | Maximum | Continuous | | | |
| | <i>R. p. m.</i> | <i>Horsepower</i> | <i>Horsepower</i> | <i>Number</i> | <i>Pounds</i> | <i>Dollars</i> |
| 1 | 1,525 | 33 | 25 | 4 | 3,100 | 1,220 |
| 2 | 1,400 | 44 | 35 | 4 | 3,500 | 1,605 |
| 3 | 850 | 60 | 45 | 3 | 5,700 | 2,120 |
| 4 | 850 | 66 | 50 | 4 | 6,300 | 2,255 |
| 5 | 850 | 80 | 60 | 4 | 6,300 | 2,470 |
| 6 | 850 | 100 | 75 | 6 | 8,400 | 3,320 |
| 7 | 850 | 125 | 95 | 6 | 8,500 | 3,495 |
| 8 | 850 | 160 | 120 | 8 | 11,500 | 5,755 |

¹ Prices for year 1940. These prices include base, clutch, special shaft, fuel tank, radiator and fan-cooling equipment, starter, and supervision of installation of engine.

Gasoline engines are made in sizes from a fraction of a horsepower up to 100 hp., and in 1-, 2-, 4-, 6-, and 8-cylinder units. Table 7 gives the cost of gasoline-power units equipped for gasoline or natural gas consumption. Most power units are truck or tractor engines that have been adapted by the manufacturer to stationary work by minor changes in design. Some automobile engines, also, have been made into power units, but in this adaptation some method of reducing the speed must be used because these engines normally operate at much higher speeds than most pumps. Furthermore, they are not designed to operate continuously at their maximum rated horsepower, and for this reason the recommendations of the manufacturer should be carefully followed when driving a pump with this type of engine. Table 8 shows the horsepower developed at different speeds by different makes of automobile engines (3). New engines should not be loaded to over 80 percent of the horsepower shown for each speed, and used engines probably should not be loaded to more than 50 to 75 percent of their rated power depending on their condition. Many slow-speed, heavy, single-cylinder engines were formerly used to drive pumps, but they are rapidly being displaced by the high-speed, lightweight power units because of the ease of operation, lower first cost, and smoother action of these units. The old engines had greater fuel economy and longer life than many of the newer types, but their disadvantages in handling a pumping load overbalanced their good qualities.

TABLE 7.—*Factory prices of gasoline-power units with accessories for burning natural gas*

| Unit | Speed | Rating of motor | | Speed | Rating of motor | | Cylinders | Weight | Price ¹ |
|--------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|---------------|---------------|--------------------|
| | | Maxi- mum | Continu- ous | | Maxi- mum | Continu- ous | | | |
| | <i>R. p. m.</i> | <i>Hp.</i> | <i>Hp.</i> | <i>R. p. m.</i> | <i>Hp.</i> | <i>Hp.</i> | <i>Number</i> | <i>Pounds</i> | <i>Dollars</i> |
| 1..... | 1, 200 | 7. 7 | 6. 2 | 1, 500 | 9. 5 | 7. 6 | 4 | 390 | 340. 00 |
| 2..... | 1, 200 | 14. 5 | 11. 6 | 1, 500 | 18. 5 | 14. 8 | 4 | 550 | 434. 50 |
| 3..... | 1, 200 | 20. 4 | 16. 3 | 1, 500 | 25. 1 | 20. 1 | 4 | 840 | 516. 00 |
| 4..... | 1, 200 | 27. 2 | 21. 7 | 1, 500 | 31. 8 | 25. 5 | 4 | 870 | 551. 00 |
| 5..... | 1, 200 | 31. 4 | 25. 1 | 1, 500 | 38. 4 | 30. 7 | 6 | 1, 140 | 709. 50 |
| 6..... | 1, 200 | 34. 0 | 27. 2 | 1, 500 | 41. 6 | 33. 3 | 6 | 1, 180 | 741. 50 |
| 7..... | 1, 200 | 43. 0 | 34. 4 | 1, 500 | 53. 0 | 42. 4 | 6 | 2, 040 | 847. 00 |
| 8..... | 1, 200 | 47. 0 | 37. 6 | 1, 500 | 57. 0 | 45. 6 | 6 | 2, 040 | 854. 00 |
| 9..... | 1, 100 | 84. 0 | 67. 2 | 1, 400 | 103. 0 | 82. 4 | 6 | 3, 360 | 1, 657. 50 |

¹ Prices are for year 1940.TABLE 8.—*Maximum horsepower ratings ¹ of new Buick, Chevrolet, and Ford engines at various speeds (3)*

| Make and year | Model | Revolutions per minute | | | |
|-------------------------------|------------------------|------------------------|------------|------------|------------|
| | | 900 | 1,200 | 1,500 | 1,800 |
| Buick: | | <i>Hp.</i> | <i>Hp.</i> | <i>Hp.</i> | <i>Hp.</i> |
| 1930..... | Series 40..... | 32 | 44 | 56 | 65 |
| 1930..... | Series 50, 60..... | 39 | 55 | 66 | 77 |
| 1931..... | Series 8-50..... | 27 | 36 | 44 | 53 |
| 1931..... | Series 8-60..... | 33 | 45 | 57 | 68 |
| 1931..... | Series 8-80, 8-90..... | 42 | 56 | 72 | 85 |
| 1932-35 ² | Series 50..... | 28 | 39 | 49 | 60 |
| 1932-35 ² | Series 60..... | 34 | 48 | 60 | 72 |
| 1932-35 ² | Series 80, 90..... | 44 | 59 | 74 | 88 |
| 1936..... | Series 40..... | 30 | 41 | 52 | 63 |
| 1936..... | Series 60, 80, 90..... | 39 | 54 | 68 | 82 |
| 1937..... | Series 40..... | 31 | 44 | 55 | 67 |
| 1937..... | Series 60, 80, 90..... | 42 | 58 | 73 | 88 |
| 1938-40 ² | Series 40..... | 32 | 45 | 57 | 69 |
| 1938-40 ² | Series 60, 80, 90..... | 43 | 60 | 76 | 92 |
| Chevrolet: | | | | | |
| 1930, 1931 ² | Passenger..... | 21 | 28 | 34 | 41 |
| 1932..... | Passenger..... | 22 | 29 | 37 | 44 |
| 1932..... | Truck..... | 22 | 30 | 36 | 44 |
| 1933-34 ² | Standard..... | 22 | 30 | 36 | 45 |
| 1933-34 ² | Master..... | 25 | 34 | 40 | 51 |
| 1933-34 ² | Truck..... | 25 | 33 | 41 | 48 |
| 1935..... | Standard..... | 26 | 35 | 43 | 52 |
| 1935..... | Master..... | 26 | 36 | 44 | 54 |
| 1935..... | Truck..... | 26 | 34 | 43 | 51 |
| 1936..... | Standard..... | 28 | 37 | 47 | 56 |
| 1937-40 ² | Master..... | 28 | 39 | 48 | 58 |
| Ford: | | | | | |
| 1928..... | Model A..... | 19 | 26 | 31 | 35 |
| 1932-33 ² | Model B-4..... | 24 | 32 | 40 | 45 |
| 1932..... | Model 18-V8..... | 22 | 30 | 38 | 44 |
| 1933-39 ² | Model 40-V8..... | 24 | 33 | 41 | 49 |
| 1940..... | 60 Hp..... | 14 | 19 | 25 | 31 |
| 1940..... | 85 Hp..... | 25 | 33 | 43 | 52 |
| 1940..... | 95 Hp..... | 29 | 39 | 50 | 60 |

¹ Buick and Ford ratings are with all accessories. Chevrolet ratings are without fan or generator which absorb from 0 to 7 percent of the power depending on the speed. Continuous load for new engines should not exceed 80 percent of maximum horsepower and not more than 50 to 75 percent for old engines depending on condition.

² Ratings of these engines may differ slightly from year to year due to minor changes of design.

ENGINE EQUIPMENT

Internal-combustion engines do not develop their rated horsepower until they are warmed up. Therefore, it is important that engines for pumping plants be equipped with clutches so that it will be possible to idle them until they approach their normal operating temperature. A clutch also makes it possible to adjust the pump without stopping the engine. The governor is another accessory that is usually included in the engine equipment. It holds the speed reasonably constant regardless of the load, and, in case the pump should lose its priming or the well go dry, it would keep the engine from speeding up; such uncontrolled speeding might seriously injure the engine unless shut off in time.

Electric-ignition internal-combustion engines are usually started by hand cranking, but Diesel engines, especially the larger sizes, must be started by mechanical means. Some Diesel engines have electric starters with heavy-duty storage batteries while others use an auxiliary gasoline engine. Some Diesel engines are equipped with spark plugs and carburetor so that they can be started as a gasoline engine. If for some reason the engine will not start, this difficulty can frequently be overcome by driving the engine with the farm tractor, if one is available. This method can also be used in starting gasoline engines.

Most power units have a radiator and fan for cooling, as standard equipment. However, since the engine can be cooled more economically by a simple coil placed in the water pumped by the plant, it usually pays to purchase the power unit without radiator or fan as there is a saving in the first cost and also in the fuel consumption because from 2 to 5 percent of the engine power is required to run the fan. Soft water should be used in the cooling system. If only hard water is available, it should not be drained except at long intervals, because each time new water is put in, the chemicals causing temporary hardness (carbonates) are deposited in the passages of the cooling system after the water becomes heated. The deposit of carbonates reduces the efficiency of the cooling system. This is also true of radiators. A valve should be placed in the cooling circuit to regulate the rate of flow through the coil which in turn regulates the temperature of the engine. When a cooling coil is used an open tank of 50 or 100 gallons in capacity should be connected into the cooling circuit in order to supply the water lost from the system by evaporation and to protect the engine against overheating in the event the pump stops discharging for any reason. Some engines are equipped with automatic devices that shut off the ignition when the engine reaches a predetermined temperature. Another type of device shuts off the ignition if the pump stops discharging.

Internal-combustion engines used to drive pumps are subjected to severe service because the load is continuous and near the capacity of the engine. For this reason, special attention should be given to the lubricating oil. Most manufacturers specify the type of oil that will give the best service in their engines and usually recommend that the oil be changed after a specified number of hours of operation. When natural gas is used a longer interval may be allowed to elapse between oil changes because it does not cause oil dilution.

Since severe dust storms frequently occur in the arid regions, it is important that the engine be equipped with an air cleaner of the oil-bath or similar type to remove the dust that would otherwise be taken into the cylinders. An oil filter of ample capacity should also be included. If filters are used and kept clean, the life of the engine will be materially increased. Breather pipes and other openings should be protected by fine screens and exposed bearings and should be equipped with felt washers or similar devices to keep out the dust.

A simple shelter for the engine and pump with provision for thorough ventilation to facilitate cooling the engine is a good investment. It protects the engine from rain and snow, and to some extent from dust. It also prevents unauthorized persons from tampering with the engine or removing parts.

PRECAUTIONS WITH DIFFERENT TYPES OF FUEL

As previously mentioned, the principal engine fuels are gasoline, kerosene, distillate, fuel oil, natural gas, manufactured gas, and butane, which is gas kept in liquid form by pressure. All these fuels should be free of foreign material and excessive amounts of sulphur and noncombustible carbon. Only fuels of the proper number of degrees Baumé³ for the particular engine should be used. This is also true of Diesel engines which will run on practically any liquid fuel, but which may be seriously injured if the fuel does not have the proper lubricating qualities. A special effort must be made to keep the fuel for Diesel engines clean because grit of any kind is very injurious to the fuel injection mechanism. Gasoline engines ordinarily will not run on any but the proper grade of fuel. The power that can be developed with natural and artificial gas is a direct function of the heating value of the gas; that is, the British thermal units⁴ per cubic foot at standard pressure. It is therefore important that the heating value of the available gas be known before an engine is purchased for operation on gas.

CHOICE OF ENGINE FOR PUMP

In choosing an engine to drive a pump, only the maximum load on the engine, which, in the case of a pumping plant, is a continuous load, need be considered. The starting load is always less than the continuous load because when the pump starts, the load increases as the water is lifted higher and higher until it reaches the outlet. After that the load is practically constant. The horsepower required can be computed by the formula:

$$\text{Horsepower} = \frac{\text{gallons per minute} \times \text{total head}}{3,960 \times \text{efficiency}}$$

or determined by reference to figure 7. The speed of the pump when delivering the required quantity of water against the total head is fixed by the design of the particular pump. This information is furnished by the pump manufacturer. Suppose that a Diesel engine is

³ Degrees Baumé indicate the density of liquid; the lighter the liquid the higher the reading. Water has a density of 10° on the Baumé scale.

⁴ A British thermal unit (B. t. u.) is the heat required to raise the temperature of 1 pound of water 1° F.

being considered and that the pump speed is 1,760 r. p. m., and the horsepower required is 31.0. An engine with a maximum of about 40 hp. should be investigated for the load because the permissible continuous loading of an engine should not exceed 75 to 80 percent of the maximum power.

The performance characteristics of a Diesel engine of about this power are shown in figure 13. This diagram indicates the maximum and rated horsepower for each speed and also the torque.⁵ The fuel consumption for various loads at the speed of 1,400 r. p. m., which is the manufacturer's recommended speed for the engine, is also given. It will be noted that the torque first increases and then decreases as the speed increases, but the horsepower continues to increase until the maximum is reached. This is because the speed increases faster than the torque decreases up to the point where friction and other losses begin to overcome the effect of the increase in speed. Beyond this point the engine should not be run because of the excessive wear. In fact, reputable manufacturers usually recommend a speed considerably below that producing the maximum horsepower because of the longer life and more satisfactory performance under these conditions.

As previously stated, the rated speed of the engine for which the performance curves are shown in figure 13 is 1,400 r. p. m. At this speed the maximum brake horsepower is 44 and the rated brake horsepower is 33. These are the points where the vertical line through the 1,400 speed intersects the maximum and rated brake horsepower curves. This engine has sufficient power to handle a pumping load of 31.0 hp., but the speed is too low for the pump. A gear head with a 4 to 3 ratio will require an engine speed of $\frac{3}{4} \times 1,760 = 1,320$. At this speed the engine has a rated horsepower of 32 which is slightly more than is required. The small change in speed from 1,400 r. p. m. to 1,320 r. p. m. can easily be made by adjusting the governor and the throttle. The total fuel consumption will be reduced in proportion to the change in horsepower because, as shown by the fuel-consumption curve, a slight reduction in the load near the maximum horsepower does not affect the fuel consumption per horsepower-hour.

If a quarter-turn V belt or a flat belt were used, the ratio of the pulley diameters would have to be as 1,320 to 1,760 with the larger pulley on the engine. If the engine pulley diameter is 12 inches, then the

pump pulley diameter is $\frac{1,320}{1,760} \times 12 = 9$ inches.

If in the previous example the computed speed had not come out so close to the speed required that the difference could be corrected by speeding up or slowing down the engine slightly with the governor, a gear head with a different speed ratio should be considered. If the pump is a horizontal centrifugal and a direct-connected unit is desired, an attempt should be made to find a pump with the required capacity that has a speed near 1,400 r. p. m.

⁵ The torque of the engine at the various speeds is the turning power of the crank shaft in pounds at a radius of 1 foot. The product of the torque, times the speed, times the circumference of the circle of radius 1 foot, divided by 33,000 is the horsepower.

DRIVES

The direct connection, the gear head, the flat belt, and the V belt are the types of drives most commonly used in transmitting power from the engine or motor to the pump. Rope and chain drives are now seldom used on pumps.

DIRECT DRIVES

The direct connection is the cheapest and most efficient type of drive and should be used whenever the pump and engine or motor speeds are identical and it is physically possible to connect the engine or motor shaft with the pump shaft. Since vertical-shaft engines are not available, it is not possible to use a direct connection on engine-driven plants when the pump is a vertical centrifugal or a deep-well turbine. If

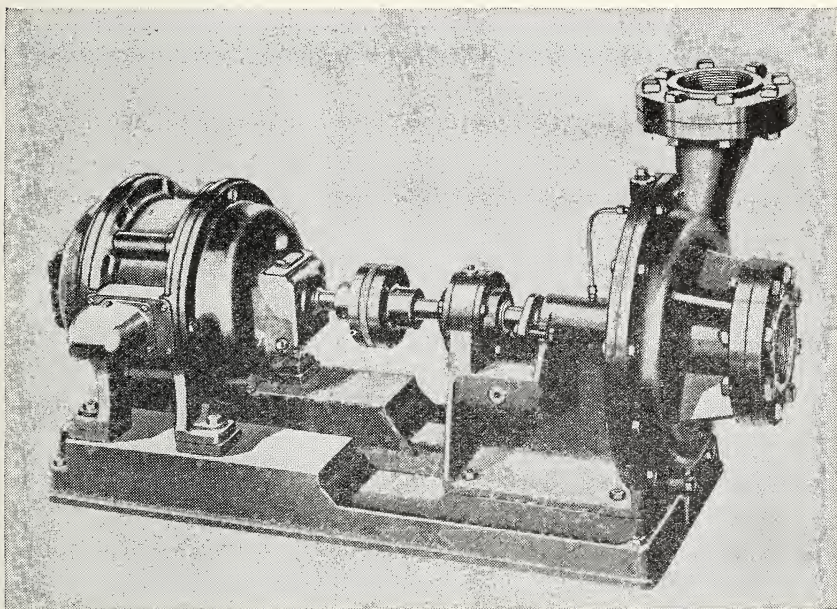


FIGURE 14.—Direct-connected motor-driven horizontal centrifugal pump.

possible the engine or motor and pump should be mounted on the same base when a direct connection is used because it is much easier to maintain correct alinement between the two shafts so mounted as to form a single unit (fig. 14). However, some type of flexible coupling should be used to connect the two shafts in order to take care of minor variations in alinement. Whether the two units are mounted on the same base or are separately mounted, the shaft alinement should be checked from time to time because errors in alinement in direct-connected units may cause a severe strain on the bearings, and, if not corrected, may cause the failure of the bearings. Excessive vibration, heating of the bearings, or any unusual noise should be investigated immediately.

GEAR HEADS

When a vertical centrifugal pump or a deep-well turbine is to be driven by an engine, the gear head is the most efficient means of transmitting the power from engine to pump (fig. 15). These units are made to fit any standard pump in place of a belted head or vertical motor. They are also made with a variety of gear ratios so that the pump and engine may each operate at their most efficient speeds. The standard gear ratios are 5 to 4, 4 to 3, 3 to 2, 2 to 1, and 1 to 1, the first number in each ratio representing the pump speed. Other gear ratios are made on special order. The gear head has a hollow vertical shaft through which the pump shaft extends. By this arrangement the pump adjusting nut and the automatic release for disengaging the pump, in case rotation is in the wrong direction, may be placed at the top of the gear head where they are readily accessible. Substantial ball bearings are provided to carry the bearing pressure and the

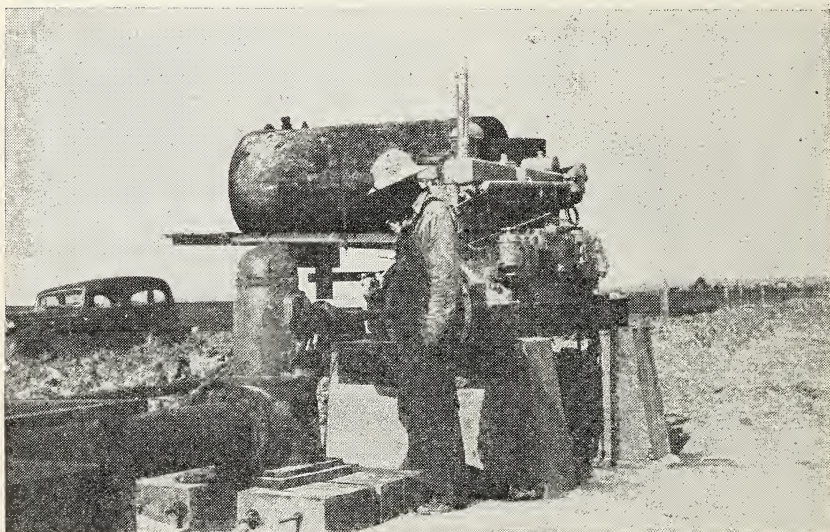


FIGURE 15.—Deep-well turbine with gear head driven by Diesel engine.

vertical thrust of the pump. The bearings and the gears are automatically lubricated and cooled when in operation. A shaft with double universal joints is recommended, to connect the engine with the gear head (fig. 16). The universal joints are provided to take care of any errors or changes in alinement of pump and engine. The distance between the pump and engine coupling should not be less than 3 feet because small errors or changes in alinement then will not materially affect the smoothness and efficiency of the power transmission. Efficiency of gear heads in transmitting power is about 95 percent.

Gear heads are sometimes used in combination with V or flat belts for the purpose of eliminating a long quarter-turn belt when it is not feasible to use a shaft connection between the units (fig. 17). Which-ever type of connection is used, it is possible to arrange the equipment so that it will occupy a minimum of space.

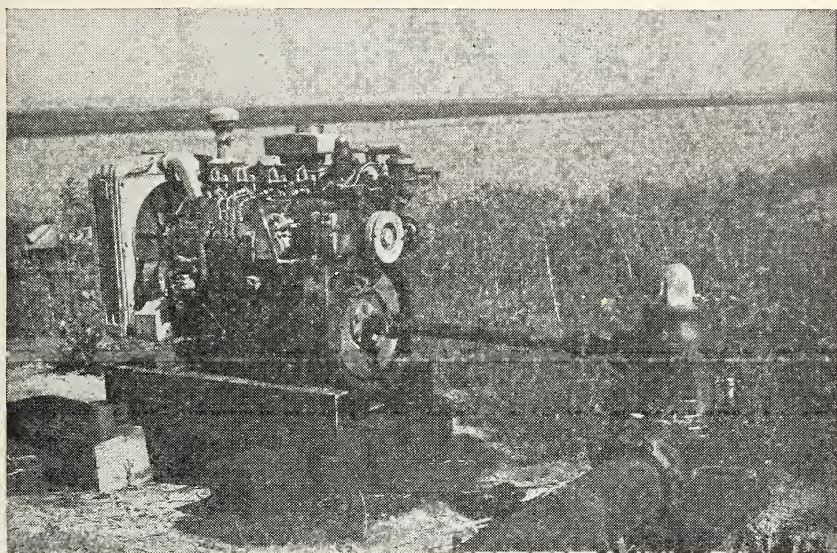


FIGURE 16.—Engine-driven pumping plant showing drive shaft with two universal joints between engine and gear head. Note that no clutch is used because auxiliary starting engine is powerful enough to start engine under initial pump load.

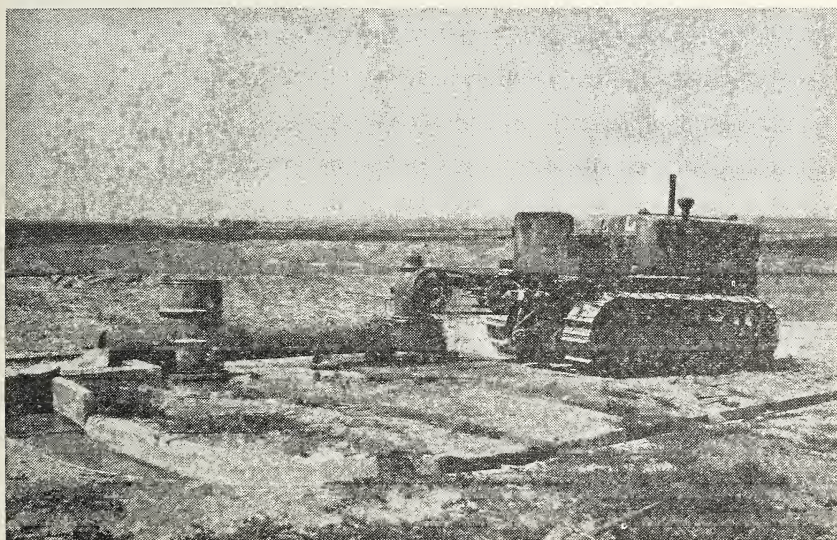


FIGURE 17.—V-belt drive on tractor-operated pumping plant. The use of a gear head on the pump eliminates the necessity for having a quarter-turn in the belt.

FLAT AND V BELTS

Flat belts have always been used extensively in driving pumping plants because of the wide variety of conditions to which they may be adapted (fig. 18). By choosing the proper pulley diameters any desired speed may be obtained. Recently V belts have come into use for similar types of installations. Flat belts are made of leather, of rubber-covered canvas, and of painted canvas. Leather and rubber belts are best for driving pumps. The width and thickness of the belt, the material of which it is made, the pulley diameters, and the belt speed determine the horsepower the belt will transmit (table 9). The data given in table 9 are for good-quality rubber belting, but they may also be used for leather belting by keeping in mind the fact that a light single-leather belt is the equivalent of a three-ply rubber belt, a medium single-leather belt is the same as a four-ply rubber belt, a heavy single-leather or a light double-leather belt is equal to a five-ply rubber belt, and a medium double-leather belt will transmit the same horsepower as a six-ply rubber belt.

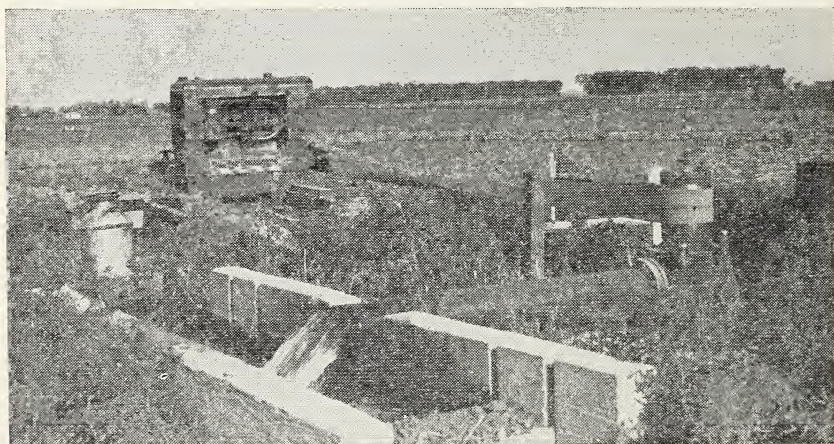


FIGURE 18.—Engine using tractor fuel driving turbine pump by means of quarter-turn flat belt.

TABLE 9.—Maximum horsepower ratings per inch of width of rubber belt¹ with 180° arc of contact²

| Plies (number) | Smaller pulley diameter | Ratings when speed of smaller pulley in r. p. m. is— | | | | | | | |
|-------------------|-------------------------------|--|-----|-------|-------|-------|-------|-------|-------|
| | | 250 | 500 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 |
| | Inches | Hp. | Hp. | Hp. | Hp. | Hp. | Hp. | Hp. | Hp. |
| 3----- | 5 | | 0.6 | 1.2 | 1.7 | 2.1 | 2.5 | 2.8 | 3.0 |
| | 6 | | .9 | 1.7 | 2.4 | 2.9 | 3.3 | 3.6 | 3.7 |
| | 8 | | 1.5 | 2.7 | 3.6 | 4.3 | 4.8 | 4.9 | |
| 4----- | 6 | | 1.0 | 1.8 | 2.5 | 3.1 | 3.5 | 3.7 | 3.9 |
| | 8 | | 1.6 | 3.0 | 4.0 | 4.7 | 5.1 | 5.1 | |
| | 10 | 1.2 | 2.3 | 4.1 | 5.3 | 6.0 | 6.2 | | |
| 5----- | 10 | 1.4 | 2.5 | 4.3 | 5.5 | 6.2 | 6.3 | | |
| | 12 | 1.8 | 3.3 | 5.6 | 7.0 | 7.6 | | | |
| | 14 | 2.2 | 4.0 | 6.7 | 8.3 | | | | |
| 6----- | 12 | 2.0 | 3.5 | 5.8 | 7.2 | 7.5 | | | |
| | 14 | 2.6 | 4.5 | 7.1 | 8.6 | | | | |
| | 16 | 3.0 | 5.3 | 8.3 | 9.8 | | | | |

¹ To find belt width, divide required horsepower by quantity indicated in the table.

² Adapted from Goodyear Handbook of Belting, ed. 2 (10).

Small pulleys should be avoided in belt drives because they cause greater belt slippage and also wear out the belts faster than large pulleys. The heavier the belt, the larger the pulleys should be. The pull should always be on the bottom belt. This causes the top belt to loosen and sag, thereby getting a better grip on the pulleys. Long belts are better than short because their angle of contact on the pulleys is increased, thereby reducing the slippage. For this reason less tension is required in a long belt, and, consequently, there is less wear on the bearings and the belt.

A flat belt will not operate satisfactorily when vertical or nearly so, because unless excessively tight, which is undesirable, it will leave the lower pulley. If it is necessary to use a belt to drive a pump in a pit, a beltway should be provided so that the belt, when installed, will run at an angle of not less than 30° with the vertical. A jack shaft is sometimes installed at the pit head to eliminate the necessity of providing a beltway, but this practice is not very satisfactory because the belt has to run vertically from the jack shaft to the pump. It does, however, facilitate tightening the belt. V belts will operate satisfactorily at a much steeper angle than flat belts, and for this reason they should be considered when driving a pump in a pit if the depth of the pit is not more than 10 or 15 feet. The recommendations of the V-belt manufacturer should be carefully followed in the designing and installation of this type.

The efficiency of flat-belt drives varies widely. A good installation may show a loss of from 5 to 10 percent in power, whereas a poor installation may show a loss of as much as 25 to 30 percent.

If a flat belt is used in driving a vertical pump, a quarter turn in the belt is required. In that event the distance between centers of the pulleys and the offset of the pulleys must be carefully determined or the belt will run off them. Figure 19 and table 10 show the proper distances and offsets. When a quarter-turn belt is used, the power transmitted per inch of belt width (table 9) should be reduced by 25 percent.

TABLE 10.—Pulley center distances and pulley offsets for quarter-turn drives, for motors and engines (14)

| Belt width (inches) | C ¹ | Motor drive | | Engine drive | |
|---------------------|----------------|----------------|-----------------|----------------|----------------|
| | | A ¹ | B ¹ | A ¹ | B ¹ |
| | <i>Inches</i> | <i>Feet</i> | <i>Inches</i> | <i>Feet</i> | <i>Inches</i> |
| 4..... | $\frac{5}{8}$ | 10 | $7\frac{1}{2}$ | 15 | 15 |
| 5..... | $\frac{7}{8}$ | 10 | $7\frac{1}{2}$ | 15 | 15 |
| 6..... | 1 | 14 | $10\frac{1}{2}$ | 22 | 22 |
| 8..... | $1\frac{3}{8}$ | 14 | $10\frac{1}{2}$ | 22 | 22 |
| 10..... | $1\frac{5}{8}$ | 18 | $13\frac{1}{2}$ | 30 | 30 |
| 12..... | 2 | 18 | $13\frac{1}{2}$ | 30 | 30 |
| 14..... | $2\frac{3}{8}$ | 24 | 18 | 36 | 36 |
| 18..... | 3 | 24 | 18 | 36 | 36 |

¹ A is distance between pulley centers; B is vertical offset of driven pulley; and C is horizontal offset of driven pulley. (See fig. 19.)

V belts transmit power more efficiently than flat belts, and they can be used under all conditions where a flat belt will serve the purpose. They will operate successfully when the pulley centers are much closer together than is permissible when flat belts are used; consequently

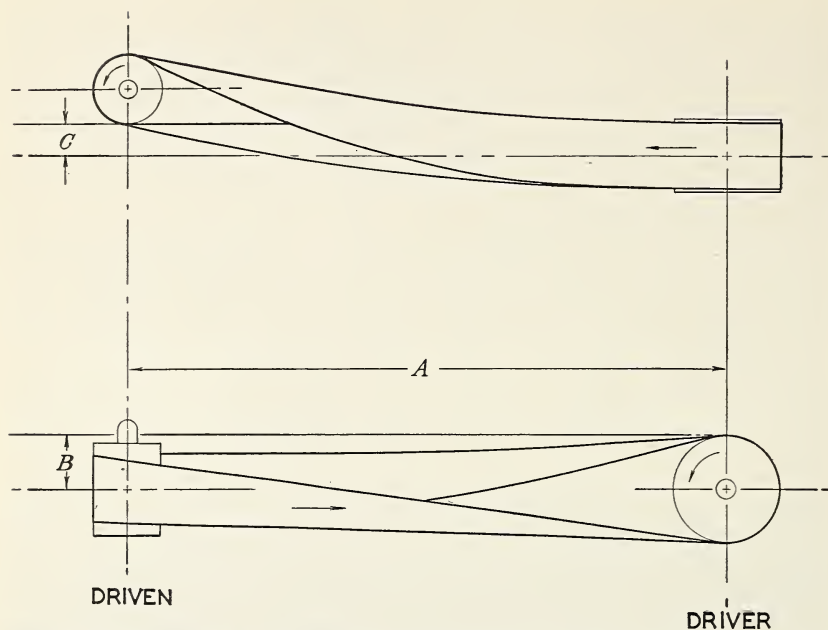


FIGURE 19.—Diagram for determining the setting of the pulleys when a quarter-turn flat-belt drive is used (14). A is distance between pulley centers, B is vertical offset of driven pulley, and C is horizontal offset of driven pulley.

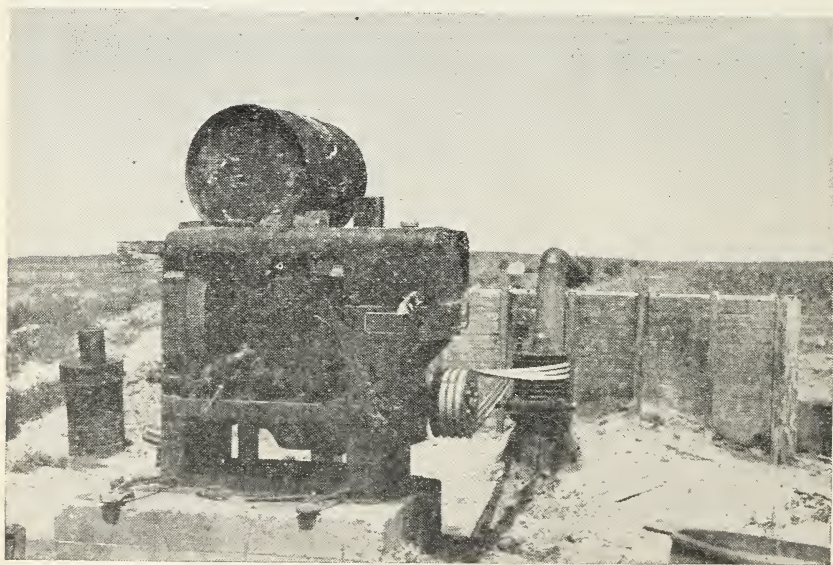


FIGURE 20.—Quarter-turn V belts being used to drive turbine pump.

they can be utilized successfully in confined spaces. Special grooved pulleys are required for V-belt drives, and since the power is transmitted by the friction of the belts in those grooves, it is important that the proper belt size be used. For this reason, when a V-belt drive is being designed, best results will be obtained if the problem is submitted to the V-belt manufacturer. V belts may also be used for quarter-turn drives (fig. 20), but they should not be used when the speed ratio is greater than 3 to 1. An efficiency of at least 90 to 95 percent is claimed for V-belt drives.

As both flat and V belts stretch with use, provision must be made for tightening the belts from time to time. The motor or engine should be mounted on a sliding base with adjusting screws so that it may be conveniently moved. A chain hoist attached to a pin driven into the ground is sometimes used as a belt tightener, but it requires frequent attention because the pin keeps bending toward the engine or motor. A heavy strut with a jack on one end between the pump and the engine or motor makes a more permanent and more satisfactory belt tightener than a chain hoist, but it is not so satisfactory as a sliding base with adjusting screws. Belts should be loosened when not in use and should be rolled and stored during the winter.

As an example of a quarter-turn flat-belt drive, assume an engine with a speed of 800 r. p. m. and a pulley diameter of 16 inches driving a deep-well turbine with a speed of 1,160. Since the speed is inversely proportional to the pulley diameter, the pump pulley diameter will have to be:

$$\frac{800}{1,160} \times 16 = 11.0 \text{ inches.}$$

Suppose that it required 32.5 hp. (engine) to drive the pump. From table 9 by interpolation 5.4 hp. appears as the power that can be transmitted per inch of width of five-ply rubber belting at a speed of 1,160 r. p. m. for a straight belt. For a quarter-turn belt, this value should be reduced 25 percent which gives 0.75×5.4 or 4.0 hp. per inch of belt. To transmit 32.5 hp. would require a belt $\frac{32.5}{4.0} = 8.1$ inches wide. Since this is only slightly greater than 8 inches, an 8-inch belt would be used.

An 8-inch belt being required, the distance between pulley centers should be approximately 22 feet (table 10 and fig. 19), and the center of the pump pulley should be 22 inches below the top of the engine pulley. The face of the pump pulley should be $1\frac{3}{8}$ inches to the right of the plane through the center of the engine pulley looking toward the pump from the engine, if the belt is turned as indicated in figure 19.

The pull should be on the bottom belt. If turning the belt as indicated in figure 19 does not give the correct direction of rotation for the pump, the belt should be turned the opposite way, but in that event the offset (*C*) should be toward the left of the center line. Before the engine or motor is started the direction of rotation of the pump should always be checked by applying current to the motor or by turning the engine over by hand. If the pump is run backwards, there is danger that the pump shaft will unscrew, making it necessary to pull the pump and perhaps causing injury to the equipment.

PIPING AND AUXILIARY EQUIPMENT

PIPING

Riveted or welded sheet-steel pipe and reconditioned standard pipe and well casing are the kinds most often used in irrigation pumping plants for the suction and discharge lines. Concrete pipe or sewer tile is sometimes chosen for discharge lines where the head is small and where there is no danger from water hammer and other shocks. Cast-iron pipe is seldom installed because of its high cost. For suction lines, standard pipe or well casing is to be preferred because it lasts longer and can be kept airtight more easily than riveted or welded sheet-metal pipe.

Sheet-steel pipe is generally made with slip joints for interconnecting the lengths. Where the pipe is under considerable pressure, lugs are welded or riveted to each side of the joints, and the joints are then held together by means of bolts. Sometimes, if the pressure is not great, the lugs are merely wired together. Threaded sleeve couplings and flanged joints are most commonly used on standard pipe and well casing. However, if the threads are battered or badly rusted, the sections of pipe may be welded together. The joints in the heavier types of sheet-steel pipe also are sometimes made by welding. Concrete pipe and sewer tile are generally cemented together. Either concrete or asphalt may be used. Of the two materials, asphalt makes the more flexible joint. Flanged joints or Dresser (Dayton) couplings should be used wherever it may be necessary to disconnect the pipe from time to time, because joints of this type can be opened without disturbing the rest of the pipe line. Unions serve the same purpose in small pipes. Dresser couplings should be installed at intervals of 200 or 300 feet in long exposed pipe lines to take care of the expansion and contraction due to changes of temperature. They are also useful in making bends in pipe lines.

Unprotected sheet-steel pipe has a life of from 5 to 15 or more years, depending on the thickness of the metal and the nature of the water and the soil. Galvanized pipe lasts longer than black pipe, and the additional cost of galvanized pipe is generally justified by the longer life of the pipe. Asphaltic dips provide some protection against corrosion, but the coating soon breaks away from the pipe. Coal-tar paints are very resistant, but they are hard to keep in place because of the tendency to flow. A recently perfected coal-tar enamel is reported to be a great improvement over the other types of paints and dips.

The carrying capacity of pipe is a function of the diameter, the head available, and the frictional resistance to flow. The friction loss depends on the kind of pipe, the diameter and the quantity of water carried. The friction loss and the velocity in riveted-steel pipe of various diameters when carrying different quantities of water are set out in table 11. The friction loss in concrete pipe is about 15 percent higher, and in steel pipe with a smooth interior about 10 percent lower, than the losses shown in the table. It is evident that the friction losses decrease in magnitude very rapidly at first as the diameter of the pipe increases, and then decrease at a much slower rate. When the point of slower decrease is reached the additional reduction in friction loss is at the expense of a disproportionate increase in pipe cost.

TABLE 11.—Loss of head due to friction, per 100 feet of new thin riveted-steel pipe, and velocity in pipe ¹

| Discharge | | Velocity in pipe in feet per second and loss of head in feet per 100 feet of pipe, according to inside diameter of pipe in inches | | | | | | | | | | | | | | | | | |
|--------------------|-----------------------|---|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|------|------|
| Gallons per minute | Cubic feet per second | 3 | | 4 | | 5 | | 6 | | 8 | | 10 | | 12 | | 14 | | 16 | |
| | | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | Ve-locity | Fric-tion | | |
| 25 | 0.056 | 1.13 | 0.20 | 0.64 | 0.05 | | | | | | | | | | | | | | |
| 50 | .111 | 2.27 | .74 | 1.28 | .18 | 0.82 | 0.06 | | | | | | | | | | | | |
| 75 | .167 | 3.40 | 1.60 | 1.92 | .39 | 1.22 | .13 | | | | | | | | | | | | |
| 100 | .223 | 4.54 | 2.76 | 2.55 | .68 | 1.63 | .22 | 1.14 | 0.09 | | | | | | | | | | |
| 150 | .334 | 6.80 | 5.96 | 3.83 | 1.46 | 2.45 | .49 | 1.70 | .20 | | | | | | | | | | |
| 200 | .446 | 9.08 | 10.31 | 5.11 | 2.53 | 3.27 | .84 | 2.27 | .35 | | | | | | | | | | |
| 300 | .668 | | | 7.66 | 5.46 | 4.90 | 1.82 | 3.40 | .74 | 1.28 | 0.08 | | | | | | | | |
| 400 | .891 | | | 10.21 | 9.42 | 6.54 | 3.15 | 4.54 | 1.29 | 1.91 | .18 | 1.23 | 0.06 | | | | | | |
| 500 | 1.114 | | | | | 8.17 | 4.81 | 5.67 | 1.97 | 2.55 | .31 | 1.63 | .10 | | | | | | |
| 600 | 1.337 | | | | | 9.80 | 6.80 | 6.81 | 2.79 | 3.83 | .68 | 2.45 | .23 | 1.70 | 0.07 | | | | |
| 700 | 1.560 | | | | | 11.44 | 9.13 | 7.95 | 3.74 | 4.47 | .91 | 2.86 | .31 | 1.99 | .13 | 1.46 | 0.06 | | |
| 800 | 1.783 | | | | | | | 9.08 | 4.81 | 5.11 | 1.18 | 3.27 | .39 | 2.27 | .16 | 1.67 | .10 | | |
| 900 | 2.005 | | | | | | | 10.21 | 6.01 | 5.74 | 1.47 | 3.68 | .50 | 2.55 | .20 | 1.88 | .12 | 1.44 | 0.05 |
| 1,000 | 2.228 | | | | | | | 11.35 | 7.35 | 6.38 | 1.80 | 4.09 | .60 | 2.84 | .25 | 2.08 | .12 | 1.60 | .06 |
| 1,250 | 2.785 | | | | | | | | | 7.98 | 2.75 | 5.11 | .92 | 3.55 | .38 | 2.60 | .18 | 1.99 | .09 |
| 1,500 | 3.342 | | | | | | | | | 9.57 | 3.88 | 6.13 | 1.30 | 4.26 | .53 | 3.12 | .25 | 2.39 | .13 |

¹ Based on Scobey formula when $K' = 0.34$ (17).

To find the most economical size of pipe for a pumping plant it is necessary to consider the total quantity of water pumped annually, the cost of pumping, the cost of pipe, and the rate of interest, taxes, and depreciation. The most economical size of pipe is the one that makes the total annual cost a minimum. In order to arrive at this cost, each problem must be investigated individually.

The amount charged to depreciation will depend on the life of the pipe. If it is assumed that the pipe will last 10 years, then 10 percent of the cost of the pipe will have to be charged to depreciation each year.⁶ A reasonable rate of interest is 5 percent, and taxes probably will not average more than 2 percent. The combined charge for depreciation, interest, and taxes will therefore approximate 17 percent annually. If the estimated life of the pipe is greater or less than 10 years, or if the interest or tax rate differs from the value given, the total to be charged for these items will, of course, be correspondingly different. The assumptions must be made to fit conditions applying to the specific plant.

A well-designed pumping plant should be capable of pumping water at a total cost of 5 cents or less per acre-foot per foot of lift. Each foot of head lost in friction will increase the cost of each acre-foot pumped by this amount. To get the total increase in cost due to pumping against the friction in the pipe, it is necessary to multiply the cost per acre-foot per foot of lift by the head lost in friction in the pipe and by the total quantity pumped. The money spent in buying larger pipe to reduce the friction loss will increase the annual cost of pumping by the amount of the depreciation, interest, and taxes on the additional expenditure. This increase in cost is borne by all the water pumped during the year.

As an example: A farmer has a pumping plant with a capacity of 900 g. p. m., and he plans to pump 200 acre-feet per annum to irrigate approximately 80 acres. A 500-foot steel pipe line is required to deliver the water at the high point of the farm. Should he purchase a 6-, 8-, 10-, or 12-inch pipe to deliver the water? It is assumed that he will be able to pump water at a total cost including fixed charges of 5 cents per acre-foot per foot of lift. It is also assumed that the life of the pipe is 10 years and that the annual interest is 5 percent and the taxes 2 percent.

According to table 11 the friction equivalent per hundred feet of 6-inch steel pipe is 6.01 feet, for 8-inch pipe it is 1.47 feet, for 10-inch pipe it is 0.50 foot, and for 12-inch pipe it is 0.20 foot. The total additional head for pipe friction for the 500 feet of pipe will therefore be 30.0, 7.4, 2.5, and 1.0 feet, respectively for the 6-, 8-, 10-, and 12-inch pipe, and the cost of the additional lift due to pipe friction, at 5 cents per foot will be \$1.50, \$0.37, \$0.125, and \$0.05 per acre-foot. For pumping 200 acre-feet the total cost of pumping against the friction head will be \$300, \$74, \$25, and \$10.

The cost (table 12) of 12-gage black steel pipe is \$0.91 per foot for 6-inch pipe, \$1.06 for 8-inch, \$1.25 for 10-inch, and \$1.47 for 12-inch. For 500 feet of 6-, 8-, 10-, and 12-inch pipe, the cost will be \$455, \$530, \$625, and \$735, respectively. The depreciation, interest, and taxes, according to the assumption made previously, will amount to

⁶ For discussion of more accurate method of computing depreciation see page 72.

17 percent per year. The annual charges against the different pipes for depreciation, interest, and taxes will therefore be 17 percent of their cost or \$77.35, \$90.10, \$106.25, and \$124.95, respectively. These costs when added to the cost of pumping the water through the different sizes of pipe are \$300.00+\$77.35, or \$377.35, for the 6-inch pipe; \$74.00+\$90.10, or \$164.10, for the 8-inch pipe; \$25.00+\$106.25, or \$131.25, for the 10-inch pipe; and \$10.00+\$124.95, or \$134.95, for the 12-inch pipe. From the foregoing computations it is apparent that the 10-inch pipe should be chosen.

TABLE 12.—*Prices of different kinds of steel pipe and cast-iron gate and check valves*¹

| Inside diameter (inches) | Price per foot of— | | | | Price per— | |
|--------------------------|--------------------|--------------------|----------------|--------------------|----------------|----------------|
| | 12-gage black | 12-gage galvanized | Standard black | Well casing, black | Gate valve | Check valve |
| | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> |
| 2..... | | | 0.19 | | 4.50 | 3.00 |
| 3..... | | | .40 | 0.35 | 7.50 | 8.00 |
| 4..... | ² 0.40 | ² 0.45 | .60 | .50 | 15.75 | 13.50 |
| 5..... | ³ .59 | ³ .63 | .82 | .66 | 22.75 | 20.00 |
| 6..... | .91 | 1.01 | 1.07 | .83 | 26.75 | 25.00 |
| 8..... | 1.06 | 1.16 | 1.47 | 1.25 | 45.00 | 46.50 |
| 10..... | 1.25 | 1.37 | 1.91 | 1.92 | 74.50 | 73.50 |
| 12..... | 1.47 | 1.61 | 2.74 | 2.65 | 103.00 | 106.00 |

¹ Prices are for the year 1940.

² 16-gage.

³ 14-gage.

If the pumping plant was required to deliver 300 acre-feet per year instead of 200, the annual charges against the pipe on account of the cost would remain the same, but the pumping costs would be increased in proportion to the amount pumped if the cost per foot of lift remained constant. Under these conditions the annual pumping charge would be \$450 for the 6-inch pipe, \$111 for the 8-inch pipe, \$37.50 for the 10-inch pipe, and \$15 for the 12-inch pipe. The total annual costs (pumping plus pipe costs), would then be \$527.35, \$201.10, \$143.75, and \$139.95 for the 6-, 8-, 10-, and 12-inch pipes. In this case the 12-inch size should be chosen.

In making these computations no consideration has been given to the fact that as the friction in the pipe decreases, the size of the motor or engine required also decreases, consequently reducing the cost of the plant and also the electric rates (in motor-driven plants). Neither has consideration been given to the fact that the efficiency of the pump will vary with the head against which it is operating or that fewer stages may be required in deep-well turbines if the friction head is reduced. In a complete analysis of a pumping problem, these factors would be considered; however, the determination of the most economical size of pipe is only an approximation, the accuracy of which is a function of the accuracy of the assumptions made as to the life of the pipe, the cost of pumping, and the interest and taxes on the line. Since these factors cannot be determined with precision in advance, a complete analysis does not seem to be warranted.

The size and thickness of the suction and column pipe of a deep-well turbine are determined by the pump manufacturer, and since the loss due to entrance, the velocity head, and the pipe friction in the suction pipe and column are charged to efficiency in a deep-well turbine, the manufacturer is usually careful to proportion the parts so that the losses will be small; otherwise, the pump will not be able to compete with one that is well designed. The thickness of the column pipe depends on the size of the pump and is based on the experience of the manufacturer. Pumps with columns of standard pipe are to be preferred. Pumps sold on a price basis, particularly when the manufacturer is reasonably sure no test will be made on the plant, should be checked to see that diameter of the suction pipe and column are not so small that there will be excessive losses caused by the high velocity in the pipes.

The suction pipe for a horizontal centrifugal pump is usually not supplied by the pump manufacturer. It is therefore necessary to design the pipe so that the losses will be kept within reasonable limits because they not only increase the head against which the pump must operate, but may also materially reduce the capacity of the pump. For example, if a horizontal centrifugal pump at an elevation of 5,000 feet above sea level has a capacity of 900 g. p. m. with a static lift of 15 feet and a suction pipe 30 feet long, it is apparent that unless the suction pipe is carefully designed, the recommended suction lift may be greatly exceeded because of the friction and other losses. However, by choosing pipe of the proper diameter these losses may be kept so small that the pump will operate satisfactorily under the conditions outlined.

For example: A pump discharging 900 g. p. m. would probably have 6-inch suction and discharge connections. If a 6-inch suction pipe were used, the friction loss per 100 feet of pipe would be 6.01 feet (table 11). Since the pipe would have an elbow where connected to the pump and some type of strainer at the inlet end to keep out the trash, additional losses would occur in these fittings. Table 13 shows the friction losses in different fittings as an equivalent length of pipe of the same size as the fitting. According to the table, the loss in the elbow is equivalent to the loss from 16 feet of pipe of the same size. Since no data are available, it is assumed that the loss in the strainer is approximately the same as that in the elbow, or 16 feet. The total length of pipe for which the friction loss must be considered is therefore $30+16+16$ or 62 feet. Since the friction loss is 6.01 feet per hundred, the total friction loss will be $6.01 \times \frac{62}{100} = 3.7$ feet. The velocity head (that is, the head required to give the water the required velocity), is equal to $\frac{V^2}{2g}$. Since the velocity, V , in a 6-inch pipe carrying 900 g. p. m. is 10.2 feet per second (table 11), $\frac{V^2}{2g} = \frac{10.2^2}{64.4} = 1.6$ feet. The total loss in a 6-inch pipe is therefore 3.7 feet (friction) plus 1.6 feet (velocity head) equals 5.4 feet. The static lift of 15 feet plus loss in pipe of 5.3 feet equals 20.4 feet which exceeds by 3.4 feet the recommended limit of suction lift of 17 feet for altitude 5,000 feet (table 14). Under these conditions

the pump would not deliver 900 g. p. m. but some smaller quantity for which the friction loss and velocity head would be enough less to bring the total lift down to about 17 feet.

TABLE 13.—*Friction in pipe fittings in terms of equivalent length, in feet, of standard pipe of the same diameter*¹

| Fitting | Friction loss in terms of equivalent length of pipe of same diameter according to inside diameter of pipe in inches | | | | | | | | |
|------------------------------------|---|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|
| | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 14 | 16 |
| | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> |
| 45° elbow | 4 | 5 | 6 | 7 | 10 | 12.5 | 15 | 17 | 18.5 |
| Long-sweep elbow | 5 | 7 | 9 | 11 | 14 | 17 | 20 | 23 | 26 |
| Standard elbow | 8 | 11 | 13 | 16 | 20 | 25 | 32 | 37 | 42 |
| Square elbow | 17 | 22 | 27 | 33 | 43 | 55 | 66 | 77 | 87 |
| Close return bend | 18 | 24 | 30 | 36 | 50 | 61 | 72 | 85 | 100 |
| Gate valve open | 1.5 | 2 | 3 | 3.5 | 4.5 | 5.5 | 7 | 8 | 9 |
| Gate valve $\frac{3}{4}$ open | 10 | 13 | 17 | 20 | 26 | 33 | 39 | 46 | 53 |
| Gate valve $\frac{1}{2}$ open | 50 | 65 | 81 | 100 | 130 | 160 | 195 | 230 | 265 |
| Gate valve $\frac{1}{4}$ open | 200 | 275 | 350 | 405 | 550 | 700 | 820 | 960 | 1,150 |
| Standard tee (through side outlet) | 17 | 22 | 27 | 33 | 43 | 55 | 66 | 77 | 87 |
| Check valves | 70 | 100 | 110 | 30 | 40 | 45 | 35 | ----- | ----- |

¹ Adapted from data given in trade publications (4, 11).

If an 8-inch suction pipe were used, the friction loss and the velocity head computed in the manner just shown would be 1.0 and 0.5 feet respectively, or a total of 1.5 feet. The static lift of 15 feet plus the loss of 1.5 feet would thus equal 16.5 feet, which is less than the recommended suction limit for the altitude. The pump would therefore operate satisfactorily under the specified conditions.

For a 10-inch pipe the friction loss would be 0.4 foot and the velocity head 0.2 foot or a total of 0.6 foot, which when added to the static lift produces a total of 15.6 feet. This size of suction pipe should be chosen if there is a possibility that the static lift may increase, because more head is available before the suction limit is exceeded.

Friction losses cannot be computed precisely because of the variations in pipe and fittings and in the flow of water and also because of the uncertainty in the laws governing friction. For these reasons the observed result may differ somewhat from the computed values.

A suction header is not recommended for pumping from a battery of wells, but if one is used the suction lines should be designed so that the friction loss from each well to the pump will be practically constant. Otherwise some of the wells will be affected more than others, and if they are shallow, the water level may be drawn down to the bottom of the suction pipe. This will break the suction in all the wells and the system will have to be primed again. Since a precise determination of the pipe diameters in order to make the friction loss the same is extremely difficult, the best way to be sure that the system will work satisfactorily is to use pipes large enough so that the friction loss will be small (1 to 2 feet). Then the increase in draw-down of the well at the pump will always be less than 2 feet by the amount of the friction loss in its drop pipe. By making the drop pipe of the well at the pump an inch less in diameter than the others a reasonably balanced system will usually be obtained.

Because of the difficulty in designing suction headers and also be-

cause pumps with suction headers are hard to prime and to keep primed, a more satisfactory plan is to use siphons to bring the water from the outlying wells to a central well in which the pump is installed. This arrangement permits using either a horizontal centrifugal pump or a deep-well turbine. Since the draw-down in the wells is reduced by the amount of the friction in the siphon, the lines should be designed to keep this loss within reasonable limits. This is easily done by use of the data in table 11. If, for example the siphon well has a capacity of 300 g. p. m. and is 60 feet away from the pumped well, and if the drop pipes in the two wells are each 25 feet long, the total length of pipe will be 110 feet. The loss per hundred feet of pipe (table 11) will be 0.74 foot for 6-inch pipe and 1.82 feet for 5-inch pipe. There will be two elbows in the siphon line, and the friction in the elbows in terms of equivalent length of pipe will be 16 feet for each 6-inch elbow and 13 feet for each 5-inch elbow. The total length of pipe to be considered in determining the friction for the 6-inch size is, therefore, $60+25+25+16+16$, or 142 feet; for the 5-inch pipe it is $60+25+25+13+13$, or 136 feet. The friction loss for the 6-inch is 1.42×0.74 , or 1.05 feet, and for the 5-inch pipe it is 1.36×1.82 , or 2.48 feet. For the conditions of the example the 6-inch pipe should be adopted because a reduction of 2.48 feet in the draw-down of the well, if 5-inch pipe were used, would materially reduce the capacity of the well. Since the velocity in the pipes is small the velocity head and the entrance loss, both of which increase the head required to drive the water through the pipe, will also be small and need not be considered. When the velocity is high these losses may amount to 1 or 2 feet.

The drop pipes should be long enough so that both the inlet and outlet will be submerged at all times, and the inlet should be submerged at least 2 feet when the water level is drawn down the maximum amount. The submergence of the outlet need not be so great, as the purpose of submergence is to seal the outlet so that air cannot get into the pipe in the event that the drop pipe flows partly full or when the limit of suction lift is exceeded by the siphon. The limiting values of the distance from the water level at the inlet of the siphon to the high point of the pipe are shown for various altitudes in table 14.

The values given in the table for siphon and pump suction lifts are the approximate practical limits of lift. If the friction in the siphon or suction line is appreciable, better performance will result if the limits of lift are correspondingly reduced.

TABLE 14.—*Theoretical suction lift, and maximum practical siphon and pump lifts at various altitudes*

| Altitude (feet) | Theoretical suction lift | Practical siphon lift | Practical pump lift | Altitude (feet) | Theoretical suction lift | Practical siphon lift | Practical pump lift |
|--------------------|--------------------------------|--------------------------|------------------------|--------------------|--------------------------------|--------------------------|------------------------|
| | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> | | <i>Feet</i> | <i>Feet</i> | <i>Feet</i> |
| 0 | 34.0 | 24.0 | 22.0 | 6,000 | 27.0 | 19.0 | 16.5 |
| 1,000 | 32.7 | 23.0 | 21.0 | 7,000 | 26.0 | 18.0 | 16.0 |
| 2,000 | 31.5 | 22.0 | 20.0 | 8,000 | 25.0 | 17.5 | 15.0 |
| 3,000 | 30.3 | 21.0 | 19.0 | 9,000 | 24.0 | 17.0 | 14.5 |
| 4,000 | 29.2 | 20.5 | 18.0 | 10,000 | 23.0 | 16.0 | 14.0 |
| 5,000 | 28.1 | 20.0 | 17.0 | | | | |

In order to be able to draw down the wells sufficiently to produce the desired quantity of water, it is usually necessary to install siphon pipes at as great a depth as possible. Frequently the trenches have to be dug down to the water table or even below it in order to keep from exceeding the limit of suction lift. Under these conditions the trench walls are likely to cave. To avoid the danger of caving and the possible loss of life or injury to workmen, the trench walls should be carefully braced. Long trenches are much more likely to cave than small pits of the same depth, and the trenches should stop several feet from the wells if they are not cased because there is danger that the well will cave if the trench it cut through into the well. A better plan is to tunnel from the end of the trench to the well; however, if very deep trenches are required to keep the siphon within the practical limit of lift it may be more economical to tunnel from one well to the other.

In laying out suction and discharge pipe lines sudden changes in pipe sizes and sharp angles in alinement should be avoided. This is especially important in suction lines. Elbows should not be attached

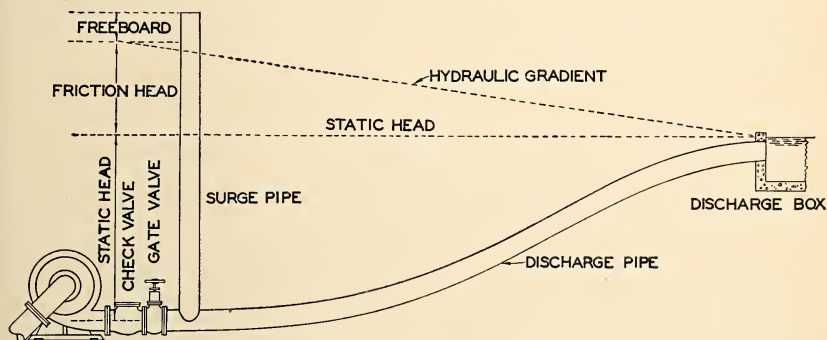


FIGURE 21.—Lay-out of pumping plant with discharge line, showing hydraulic gradient and check valve, gate valve, and surge pipe for the reduction of water hammer. Minor changes in hydraulic gradient at valves and surge pipe not shown. (Not to scale.)

directly to the inlet of the pump because when this is done the entrance conditions will be unsatisfactory. A short section of pipe should be placed between the elbow and the pump inlet. This will help materially in straightening out the lines of flow before the water enters the eye of the impeller. It is also important that no part of the suction pipe be higher than the inlet to the pump because the suction line operates under a partial vacuum, and if there is a high point in the line, air will tend to accumulate at this point.

Since discharge lines are under pressure except at the outlet, changes in grade are permissible unless they cause the line to rise above the hydraulic gradient. When this occurs the lines will be under a partial vacuum and air will accumulate in this line at the high point, as in the case of the suction lines. Whether the discharge line extends above the hydraulic gradient can readily be determined by plotting a profile of the pipe line on any convenient scale, as shown in figure 21. Then from a point directly over the pump outlet at the level of the point of discharge of the pipe line,

a line should be drawn vertically upwards, equal in length to the indicated friction in the pipe and fittings as determined from tables 11 and 13, and on the same scale as the vertical scale of the pipe profile. From the top of this line a straight line should then be drawn to the outlet of the discharge pipe. If any part of the pipe is above this line it should be lowered until it is below the line. This can be done by deepening the cut in the section where the pipe is above the hydraulic gradient. If the pipe line is discharging under pressure (as when sprinklers are used), both ends of the hydraulic grade line should be raised by the amount of the pressure in feet. When the pipe is discharging under pressure there is little likelihood that it will extend above the hydraulic gradient unless the ground surface is very rough.

Air vents should be provided at all high points in the discharge line because air or other gases in the water will accumulate there if no outlet is provided. Air-relief valves should be used where there is considerable pressure on the line, but where the high points are near the hydraulic gradient and the pressures are consequently low, it is cheaper and probably more satisfactory to install stand pipes. The pipes should extend a foot or more above the hydraulic gradient. For most installations, 1¼-inch pipes are large enough.

METHODS OF PRIMING PUMPS AND SIPHONS

One of the most satisfactory methods of priming horizontal centrifugal pumps and siphons of small plants is by means of a pitcher pump. In priming a siphon, the pump should be attached at the high point of the line, and in priming a pump, at the top of the case. Since both ends of siphons should be submerged, no valves are necessary in the siphon, but a small globe valve in good condition should be attached below the pitcher pump in order to shut it off after the siphon is primed. In order to prime a horizontal centrifugal pump, however, provision must be made for closing the discharge pipe. This can be done best by installing a gate valve in the discharge line at the pump. As a measure of economy the gate valve should be the same diameter as the pump outlet rather than the discharge line, which is usually made larger in order to reduce the friction. The increase in friction caused by the use of the smaller valve is negligible. Closing the outlet of the discharge line by means of a carefully fitted plug makes it possible to prime the pump, but this method should be used only in an emergency.

It is, of course, essential that all fittings and valves be airtight. Also, the higher the lift and the greater the volume of air that has to be removed by the priming pump, the longer it will take to prime the pump or siphon. When there are several siphon lines to prime as well as the pump, a double-acting pump of the horizontal cylinder type having a larger capacity than a pitcher pump will save time. By means of suitable piping and valves one pump can be made to serve all the needs. For most satisfactory service the pump should be firmly mounted in the pump house where it will be protected from the weather. Drain plugs should be removed from the pump at the end of the irrigation season in cold climates in order to avoid damage by freezing.

As soon as the pump or siphon is primed, the valve on the line to the priming pump should be closed, and when all the lines are primed, the pump should be started. When the pump is up to speed, the discharge valve should be opened. As soon as the water in the pumped well begins to lower, the siphons from the other wells will start to flow, and the flow will increase until the pumped well is drawn down to the limit or the suction limit of the siphon is reached.

CHECK VALVES AND FOOT VALVES

Whenever a pump discharges into a long discharge line, some means must be provided for automatically closing the discharge line when the pump is shut off or stops, so that all the water in the pipe will not drain back into the well. The most satisfactory means is a swing check valve in the discharge line near the pump (fig. 22). When the direction of flow of the water reverses, the hinged gate in the check valve swings back against its seat and closes the line. Check valves of the type usually sold are expensive because they are made for high

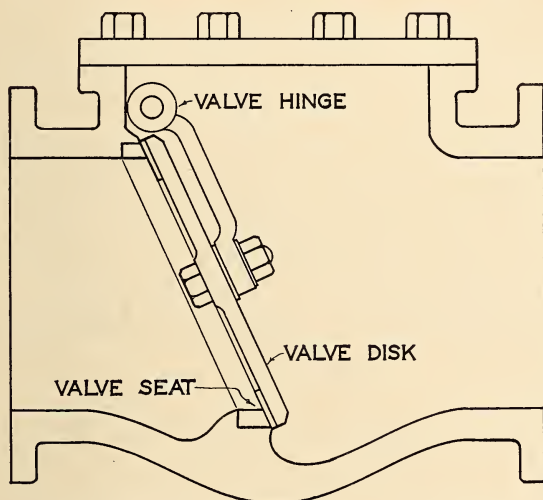


FIGURE 22.—Check valve for use in long discharge pipe where the water hammer may be considerable.

pressure service. A satisfactory check valve for low pressure service can be made by an experienced welder. Second-hand check valves may be purchased at a considerable saving.

Check valves cause resistance to the flow of water, which must be considered in determining the total head against which the pump will operate. Approximate values of the loss are given in table 13.

Some manufacturers recommend the use of foot valves on the suction line of horizontal centrifugal pumps because the foot valve prevents the water in the discharge line from draining out when the pump stops, thereby eliminating the necessity of priming the pump when next started. Actually, however, this desirable condition is seldom realized for the reason that the foot valve almost always fails to close completely because small rocks or sticks get under the seat. As a

result, the water leaks out and unless the pump is started within a short time other means must be provided to prime it. Since pumps with foot valves seldom have gate valves or check valves in the discharge line, it is necessary to disconnect the suction pipe in order that the valve may be inspected and cleaned. Performance of this task is usually difficult. If the discharge line can be plugged, it will be possible to prime the pump with a pitcher pump, as previously explained. If the leak in the foot valve is small, it may be possible to pour enough water into the discharge pipe to bring the water level over the pump impeller, which is sufficient for starting the pump. Foot valves increase the loss of head in the suction line where it is especially undesirable. For this and the reasons previously mentioned, it is believed that foot valves should not be used generally. No information is available as to the loss of head through foot valves.

SURGE PIPES

Check valves and foot valves close suddenly when the direction of flow in the pipe reverses. If the pump is discharging through a long pipe line, say 500 feet or more, the sudden closing of these valves will cause a great increase in pressure in the pipe because the long column of water has to be stopped the instant the valve closes. This sudden stoppage of the flow causes what is known as water hammer. The increase in pressure in feet caused by suddenly stopping the flow in the pipe may have a maximum value as high as 130 times the velocity of flow at the time the valve closes (20). If no outlet for the water is provided, the high pressure built up may injure the pump or rupture the pipe. The simplest method of protecting the pump and pipe line against this pressure is to install a surge pipe near the pump.

The surge pipe is a vertical pipe direct-connected to the discharge line and of a height somewhat greater than the total pressure in feet at the point where the pump is located; that is, the static pressure plus the friction and other losses (fig. 23). The excess height is required because there is a certain amount of surge in the line caused by starting the pump. If the head against which the pump is operating is more than 20 feet, a surge pipe is not so satisfactory because it is too tall to stand without additional support. Where this occurs an air chamber or relief valve should be used. These devices will be described later (pp. 56-59).

When a discharge line is equipped with a surge pipe, the pressure built up by the sudden closure of a check valve or foot valve will cause water to flow up into the surge pipe until the pressure is relieved. If a high velocity is developed before the valve closes, the pressure built up by the closing of the valve may force the water out of the top of the surge pipe. This will cause no harm. Care should be exercised, however, to have the surge pipe far enough away from the pump so that water will not splash on the belt if one is used.

Since the maximum pressure increase in feet due to sudden stoppage or flow in a closed pipe may be 130 times the velocity at the time the valve closes, the check valve should be properly designed and the parts should move freely so that the valve will close the instant the water stops flowing after the pump is shut off and before the water can start to flow back into the well. At this instant, the velocity

is nearly zero; consequently the pressure built up by the closing of the check valve will be very small. If, however, the valve hinge does not operate freely, the reverse flow may have to develop a high velocity before it is strong enough to close the valve. When this occurs a pressure equal to several hundred feet of head may easily be developed. It is therefore important that freely closing check and foot valves be used on installations of this sort.

When a surge pipe is used, the high pressures that may be developed by the sudden closing of a foot or check valve are dissipated in forcing water up into the pipe until the pressure is relieved. The amount of

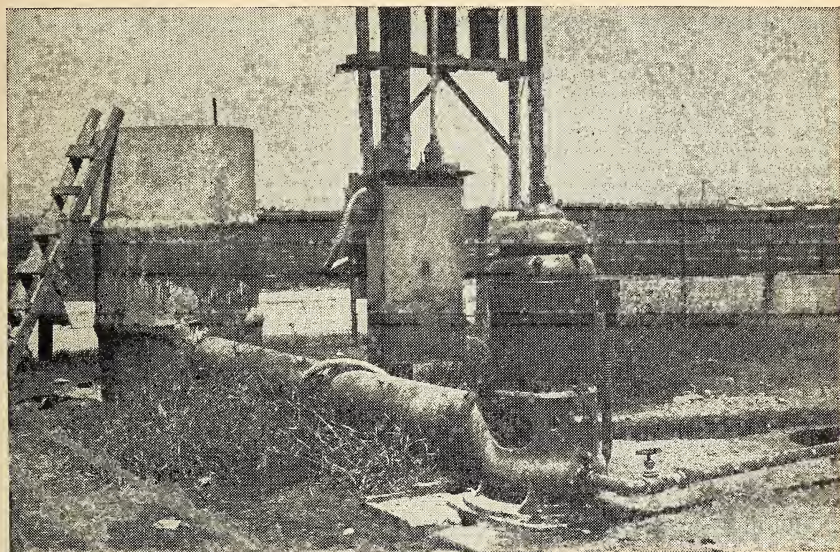


FIGURE 23.—Concrete surge pipe on discharge line. This type of surge pipe also serves as a gate chamber.

rise of the water in the surge pipe is a function of the length of the pipe and the velocity of flow at the time the valve closes, and the ratio of the area of the discharge line to the surge pipe. It is expressed by the formula ⁷ (21)

$$S = V \sqrt{\frac{L}{g} \frac{P}{A}}$$

where S is the rise in the surge pipe, in feet

V is the velocity in the pipe, in feet per second, at the time the valve closes

L is the length of the pipe, in feet

g is 32.2 feet per second per second, the acceleration due to gravity

P is the area of the pipe, in square feet, and

A is the area of the surge pipe, in square feet.

The rise S is measured from the level of the water in the surge pipe at the time the valve closes. This level will be below the stage existing

⁷ This formula disregards pipe friction, compression of water, and expansion of pipe.

when the pump is discharging, the difference depending on how soon the valve closes after the direction of flow in the discharge line reverses. This is one of the reasons why more water may splash out of the surge pipe when the pump starts than when the check valve closes. If the surge pipe and the pipe line are the same diameter

$$S = V \sqrt{\frac{L}{g}}$$

that is, the rise S in the surge pipe is independent of the pipe diameter in this case.

If, for example, a pipe line carrying 900 g. p. m. is 10 inches in diameter and 400 feet long and has its outlet 15 feet above the pump, the height of the surge pipe should be 15 feet plus the friction in the pipe plus a freeboard of say 5 feet. The friction in the pipe (see table 11) will be 0.50 foot per hundred feet or 2.0 feet for 400 feet. Friction in the check valve is not included because it is between the surge pipe and the pump. Therefore, the height of the surge pipe should be $15 + 2 + 5$, or 22 feet. If no surge pipes were provided and the check valve should close when the reverse velocity reached 0.5 foot per second, the total pressure head developed by the sudden closing of the valve would be 130 times 0.5, or 65 feet. If the velocity attained were 1 foot per second, the head developed would be 130 feet.

If a surge pipe of the same diameter as the pipe were installed, the rise S in the surge pipe due to the sudden closing of the check valve

would be equal to $V \sqrt{\frac{L}{g}}$ or $0.5 \sqrt{\frac{400}{32.2}} = 0.5 \times 3.53 = 1.76$ feet if the

velocity at the time of closing was 0.5 foot, 3.53 feet if the velocity was 1 foot and 35.3 if the velocity attained 10 feet per second before the valve closed. From the foregoing examples it is evident that if the check valve closes within a reasonable time, the rise in the surge pipe will not be sufficient to cause the water to be forced above its top if it is tall enough for the normal operation of the plant. If, however, the water running back into the well should reach a high velocity before being shut off by the valve, as in the last example, a great surge of water would shoot from the top of the pipe.

AIR CHAMBERS

Since a surge pipe is not practicable for protecting the pipe and equipment against water hammer if the rise of the discharge line is more than 20 feet, an air chamber or a relief valve should be used in these circumstances. Air chambers for long pipe lines must have greater capacity than those frequently seen on small pumps. The customary practice is to install a length of pipe with sealed end of the same diameter or larger than the discharge line, in the same manner as a surge pipe, except that the connection between discharge line and air chamber is usually made with smaller pipe (fig. 24). To serve its purpose the chamber must be filled with air rather than water, and means must be provided for determining the amount of air. This is usually done by providing petcocks at suitable intervals. Gage glasses, although easily broken, are sometimes used.

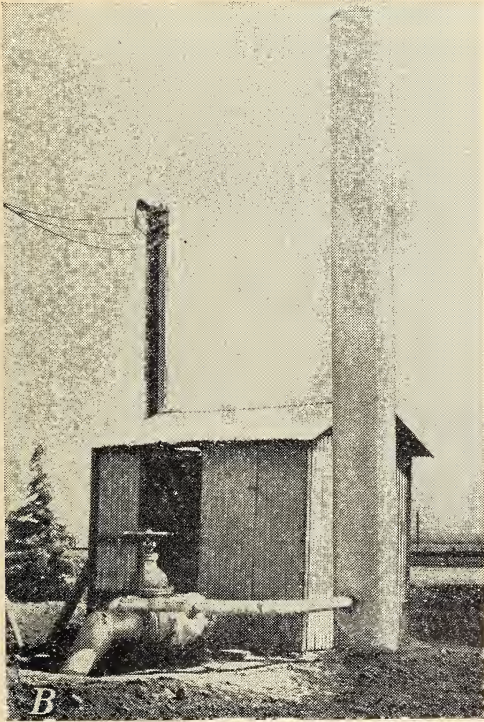


FIGURE 24.—Air chambers on discharge lines for reducing water hammer: *A*, Air chamber direct-connected to discharge line by means of vertical pipe welded into line. Flexible coupling, check valve, and gate valve are also shown. *B*, Air chamber with independent support and with horizontal pipe connecting discharge line and air chamber. Note that the connection is made to top of discharge line so that each time the pump is started air will be forced into the air chamber.

The height and diameter of the air chamber will depend on the length of pipe line and the velocity of the water when the check valve closes. For short pipe lines where the pressure head during pumping does not exceed 20 feet an air chamber 5 or 6 feet high and the same diameter or larger than the pipe line will usually be satisfactory. For long pipe lines and high heads, larger air chambers must be provided. The precise determination of the size is difficult because there are so many uncertain factors involved. Under these circumstances an engineer should be employed or the problem should be submitted to the engineers of the pump or pipe company that supplied the equipment, as they usually have had experience with problems of this sort.

An approximate formula ⁸ for determining the size of air chamber necessary to keep the maximum water hammer pressure below a predetermined value, given by Dawson and Kalinske (5), is as follows:

$$U = \frac{LAV^2}{2gH^1 \left(2.3 \log \frac{H_m^1}{H^1} + \frac{H^1}{H_m^1} - 1 \right)}$$

in which U = air chamber volume, in cubic feet, when check valve closes.

L = length of pipe, in feet.

A = area of pipe, in square feet.

V = velocity of flow in pipe, in feet, when check valve closes.

g = 32.2 feet per second per second acceleration due to gravity.

H = pressure, in feet, of water at air chamber when valve closes.

H^1 = absolute pressure in feet of water at air chamber when valve closes, = H + atmospheric pressure = H + 34 feet at sea level.

H_m = maximum pressure, in feet, of water due to water hammer.

H_m^1 = H_m + atmospheric pressure, = H_m + 34 feet.

To solve the formula for U it is necessary to adopt values for V , H , and H_m ; L and A are fixed by the existing conditions. The value of V , the velocity in the pipe at the time the check valve closes, will depend on the length of the interval between the time the water starts flowing back into the well and the time the valve closes. Since U is a function of V^2 it is obvious that the check valve should close as soon as possible after the reversal of flow begins in order that V may be kept as small as possible. It is assumed that most check valves will close before the velocity reaches 2 feet per second. H , the pressure at the air chamber at the time the check valve closes, depends on the characteristics of the pipe line, the pump, and the motor or engine. Unless the actual value of H at the time the check valve closes is known, it will probably be sufficiently accurate, under ordinary conditions, to adopt a value equal

⁸ This formula does not take account of compression of the water or expansion of the pipe.

to one-half the static pressure. Close determination of this pressure is not important because the absolute pressure is used in the formula; consequently, an error of 1 or 2 feet is only a small percentage of the total. H_m , the maximum pressure allowable, depends on the size and thickness of the pipe and the material.

As an example of the use of the formula, take a pipe 1,000 feet long and 12 inches in diameter with a maximum allowable pressure of 50 feet. It is assumed that the check valve will close when the velocity V reaches 2 feet per second and that at this time the pressure H is 5 feet. For conditions at sea level H^1 is then equal to $H+34$, or 39 feet, and H_m^1 is equal to $50+34=84$ feet. (For other altitudes, see table 14 for value of atmospheric pressure.) Substituting these values in the formula gives

$$U = \frac{1,000(0.7854)2^2}{64.4(39)\left(2.3 \log \frac{84}{39} + \frac{39}{84} - 1\right)} = 5.4 \text{ cubic feet,}$$

the volume of air required in the air chamber.

If the pressure H had been 7 feet instead of 5, the volume of air required as determined by the formula would have been 5.8 cubic feet. In other words, a 40-percent change in H would change the value U by a little more than 7 percent, which is well within the limit of accuracy expected in water hammer problems.

When the volume of air required has been determined, the length of the air chamber required for any diameter of pipe can be computed easily. It makes no difference in the results whether the pipe be short with large diameter or long with small diameter, except that if the diameter is large the wall of the pipe must be thicker.

The air in the air chamber is gradually absorbed by the water because of the increased pressure, and for this reason the amount of air in the surge chamber should be checked from time to time by tapping the pipe. The approximate height determined in this manner can be checked by opening one of the petcocks near the dividing line between the water and air, preferably where there is water, because a large amount of air will be lost if the petcock is opened into the part where the air is held under high pressure. In no event should there be any fittings in the upper half of the air chamber because there is always danger of a leak, and even though very small it would permit all the air to escape in the course of several days on account of the high pressure. In the event the air chamber is found to be full of water, the pump should not be stopped until sufficient air has been pumped into the chamber to protect the pipe. Starting and stopping the pump usually supplies sufficient air for the air chamber on lines where the pressure is light. On high-pressure lines an auxiliary source of air will probably have to be provided.

Relief valves are lever- or spring-controlled valves that open when the pressure in the pipe line reaches a predetermined value. This allows water to escape, which relieves the pressure when the check valve closes. These types of valves have not been found satisfactory in field installations because they usually leak and may stick when they are supposed to function. They are also expensive.

Discharge lines containing inverted siphons should be provided with waste valves at the lowest points for draining the line. This is of particular importance where there is danger that frost will burst the pipe. To insure its complete drainage, air valves should be installed at the high points in the line unless there are turn-outs at these points.

When there are several outlets on the discharge pipe, all but the highest outlet must be equipped with gates. The ordinary slide gates, such as those used in irrigation canals, are not suitable for this service because they leak too much. Gate valves may be used, but they are too expensive for most installations. The most satisfactory and cheapest gates for this service where the head does not exceed 50

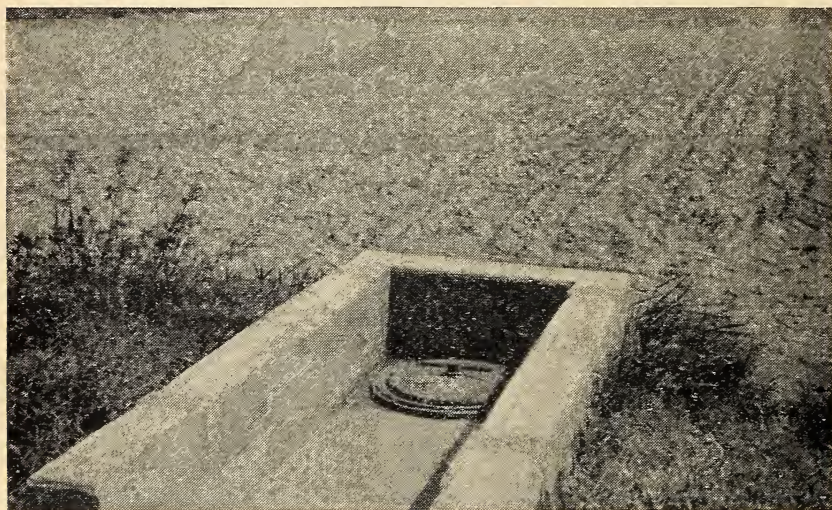


FIGURE 25.—Alfalfa valve in pit. This arrangement reduces the splash when valve is opening or closing.

feet are those of the alfalfa-valve type shown in figure 25. These valves are made so they may be easily attached to a short riser on the discharge pipe. The principal objection to this type of gate is that it splashes considerably when opened or closed unless it is set about 18 inches down below the ground level. However, such an installation reduces the erosion of the canal banks near the outlet. The prices of alfalfa gates are given in table 15.

TABLE 15.—*Cost of alfalfa valves at factory*

| Size of pipe (inches) | Size of opening | Weight | Price ¹ | Size of pipe (inches) | Size of opening | Weight | Price ¹ |
|--------------------------|--------------------|---------------|--------------------|--------------------------|--------------------|---------------|--------------------|
| | <i>Inches</i> | <i>Pounds</i> | <i>Dollars</i> | | <i>Inches</i> | <i>Pounds</i> | <i>Dollars</i> |
| 6..... | 6 | 16 | 5.95 | 12..... | 12 | 38 | 11.90 |
| 8..... | 8 | 27 | 8.15 | 14..... | 14 | 48 | 13.85 |
| 10..... | 10 | 30 | 9.90 | 16..... | 16 | 59 | 17.40 |

¹ Prices are for year 1940.

BIDS AND PURCHASE AGREEMENTS

Before purchasing a pump a farmer should investigate the equipment offered by several reputable dealers or manufacturers to find out which company has the equipment most nearly suited to his needs. In making his selection he should not be governed by price alone because the cheapest pump is perhaps not well made and usually is not adapted to his conditions. Unless he has had considerable experience he should discuss the matter with some disinterested person who is familiar with pumps. The foregoing statement applies equally to the purchase of engines.

Usually it is not necessary to call for formal bids when purchasing the equipment for a pumping plant unless it be a large one. However, the equipment for even a small plant may be quite expensive if the depth to water is great. Where this is the case, formal bids should be obtained and careful specifications drawn in order that each company will bid on equipment for the same conditions. The principal terms the bidders will have to know are depth to static water level, quantity of water to be pumped, the drawn-down, the fluctuation of the water level, diameter of the well, and the types of power available. If electric power is to be used, the voltage and number of cycles of the current should be given. The specifications should state whether the price is to include installation of equipment, and whether wiring is included. During the growing season it is important also to specify the time of delivery.

The best practice is to invite bids for a completed plant ready to deliver water. This arrangement puts all the responsibility in one place. Then if the plant does not function satisfactorily, the manufacturers of the different parts of the plant cannot throw the blame for the failure on the manufacturer of some other part. When one agency supplies the completed plant, if any part is unsatisfactory it is the duty of the contracting agency to determine what part of the plant is at fault and make arrangements for correcting it. Furthermore, agencies supplying pumping plant equipment are better able to deal with manufacturers of unsatisfactory equipment than the farmer.

Most pumps and pumping equipment are purchased as the result of dealings with pump salesmen who are the representatives of pump companies or machinery supply houses. These men generally have specialized in selling pumping machinery and usually are able to advise the farmer correctly as to the type of equipment he should buy. However, not all salesmen are of this type, and for this reason the farmer should be on his guard, especially in regard to extravagant claims.

After the farmer has decided on the make of pump he intends to purchase, a purchase agreement should be drawn up. This is a contract between the farmer and the pump machinery supply company covering the delivery of specified equipment to the farmer. The contract should state the nature of the different pieces of equipment to be delivered, and it should describe them briefly by make and rating. The terms of the contract in regard to the installation of the pump, motor, starter, and wiring should be fully covered. The time of delivery and installation should be mentioned, as should also the

method and date of payment for the equipment. The contract should stipulate whether the vendor or the purchaser will pay the freight and who will pay the living expenses of the erecting crew if one is provided by the vendor. The contract should promise also the delivery of the pump performance curves to the farmer.

The most important part of the contract, however, is the performance guarantee. This guarantee should contain the following information: The maximum horsepower required at the meter for direct-connected units or at the pulley for belt-driven units when a specified quantity of water is being pumped against a specified head as defined below. The same information should be given for the pump when operating against a head 10 percent greater and 10 percent smaller than the normal head. Under special conditions, it may be desirable to specify other heads for which the quantity pumped and horsepower required should be included in the contract.

The pumping head for a deep-well turbine is the vertical distance from the water level in the well while pumping is at the specified rate, to the center of the discharge elbow of the pump, plus the total head above this point. In other words, all head losses between the level of the water in the well and the center of the discharge elbow are not part of the pumping head as interpreted in the contract, but the static head and all losses in head beyond the pump elbow are a part of the head. This interpretation of the pumping head has been generally adopted because it simplifies test procedure and also eliminates many sources of controversies.

The pumping head for a horizontal centrifugal pump is the sum of the difference in elevation between the water surface in the well when pumping is at the specified rate and the center line of the pump, plus all losses between these points except those in the pump, plus the total head on the discharge side of the pump measured from the center line of the pump. The losses included in the pumping head for a horizontal centrifugal pump that are not included in the pumping head for a deep-well turbine are the loss of head at entrance, the velocity head, and the friction in the suction pipe and fittings.

Under the terms of the contract, the capacity and horsepower may vary from the guaranteed quantities by not to exceed 5 percent. This provision is included in the contract because the unavoidable errors in field tests may easily introduce an accumulated error of this magnitude into the results.

Most pump companies have their own contract form, but if one of this type is used the printed matter on the back of the form should be carefully read before the contract is signed. A form of contract approved by various agencies has been prepared which is fair to both the vendor and the purchaser. Copies of this contract form may be obtained by writing to the Division of Irrigation, Soil Conservation Service, Box 180, Berkeley, Calif.

The contract for the purchase of an engine should specify the items to be included because the accessories needed depend on the nature of the service to which the engine is to be put. Items that should be kept in mind when the contract is being prepared are governor, air cleaner, oil filter, fuel tank, radiator, clutch, starting equipment, extension shaft pulley, and engine base. The price will depend on the items included, and those not needed for the specific installation

should be excluded. The contract should show the fuel consumption, the maximum brake horsepower and the rated horsepower for continuous operation at the operating speed. The same data should also be given for specified speeds above and below the operating speed, depending on the possible range of conditions to be met. A chart showing the characteristics of the engines of the series purchased and a copy of the engine specifications should be included in the items to be delivered.

Whether the purchase involves a pump or an engine, it should be kept in mind that the reputation of the manufacturer for fair dealing and the production of efficient and dependable equipment is as important as the terms of the contract. The type of service rendered by the manufacturer in repairing equipment and supplying parts is also important. The contract can only guarantee the delivery of the equipment specified; whether it will give satisfactory service depends on the manufacturer.

TESTING PUMPS, ENGINES, AND MOTORS

The test of a small pumping plant should always be made in the presence of the owner and the vendor or their representatives. The test should include the determination of the head, the quantity pumped, and the power consumed. In the determination of the head, the method used will depend on whether the pump tested is a horizontal centrifugal pump or a deep-well turbine. The heads to be measured in each case are described on page 15.

When the suction head on a centrifugal pump is being measured, water or mercury manometers should be used because they are more accurate than vacuum gages for field use. In measuring the head on the discharge side of the pump a pressure gage may be used for heads over 20 feet, but for smaller heads a mercury or water manometer will give more accurate results. Water manometers should be used for the smallest heads. The gages should be connected at the center lines of the suction and discharge pipes and perpendicular to them. It is important that the holes through the pipes be smooth on the inside and that no part of the gage connection extend into the pipe. If the discharge pipe is at a higher level than the suction pipe, the difference in elevation between the gage connections should be added to the manometer readings to get the total head, and if the discharge pipe is lower, the difference in elevation between the gage connections should be subtracted from the total head. In case the suction and discharge lines are not the same diameter, the total head should be increased by the difference in the velocity head⁹ of the two pipes if the discharge line is smaller than the suction pipe and reduced by the difference in velocity head if the suction line is smaller. This correction is usually disregarded because it is small in a well-designed plant.

⁹ Velocity head $h = \frac{V^2}{2g} = \frac{V^2}{64.4}$, where V is the velocity in the pipes, in feet per second.

The total head may be determined also by measuring the difference in water levels and computing the friction and other losses. When this method is used, the velocity head of the discharge should be added to the total head. Figure 26 shows the arrangement of the gages for determining the head (12).

When determining the head for a deep-well turbine, the distance between the center line of the discharge elbow and the water surface in the well may be measured with a tape if the well is large and relatively shallow. Otherwise, an electric sounder or an air line and pressure gage (fig. 27) should be used. The air line should extend at least 5 feet below the suction inlet or should end 5 feet above it because the velocity of the water near the inlet might affect the results. The lower

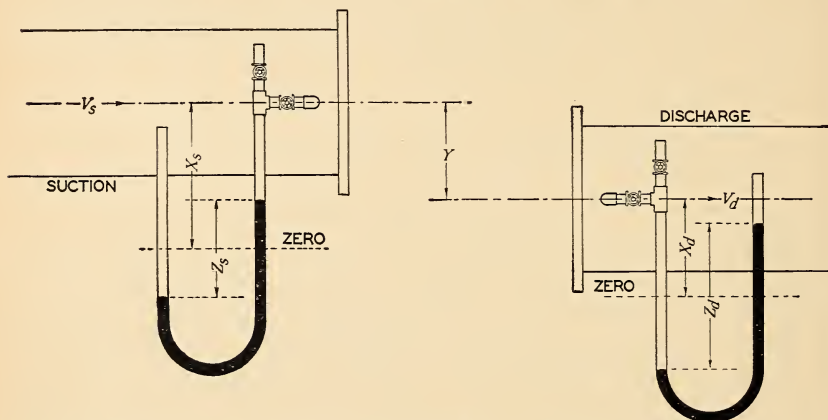


FIGURE 26.—Arrangement of mercury gages for measuring head on suction and discharge lines: Suction head $H_s = 13.596Z_s + X_s - \frac{1}{2}Z_s$; discharge head $H_d = 13.596Z_d - X_d - \frac{1}{2}Z_d$; total head $H = 13.596(Z_s + Z_d) - \frac{1}{2}(Z_s + Z_d) + (X_s - X_d) - Y + \frac{V_d^2 - V_s^2}{2g}$. H_s , H_d , Z_s , Z_d , X_s , and X_d are measured in feet and V_s and V_d , the pipe velocities, in feet per second. Connections between gages and pipe lines must be kept filled with water.

end of the air line must always be submerged. To determine the water depth by this method the length of the air line must be known. By pumping air into the line the pressure is built up and water is forced out of the line. This continues until all the water is forced out and the pressure in the line just balances the head produced by the water on the outside of the air line. Pumping additional air into the line will not build up the pressure because the air will escape from the bottom of the pipe. The lift is equal to the length of air line, in feet, from the center of the discharge elbow minus pressure on the line, in feet. If the gage reads in pounds, the conversion to feet is made by multiplying the reading by 2.31; and if the gage reading is in inches of mercury, it is made by multiplying by 1.13.

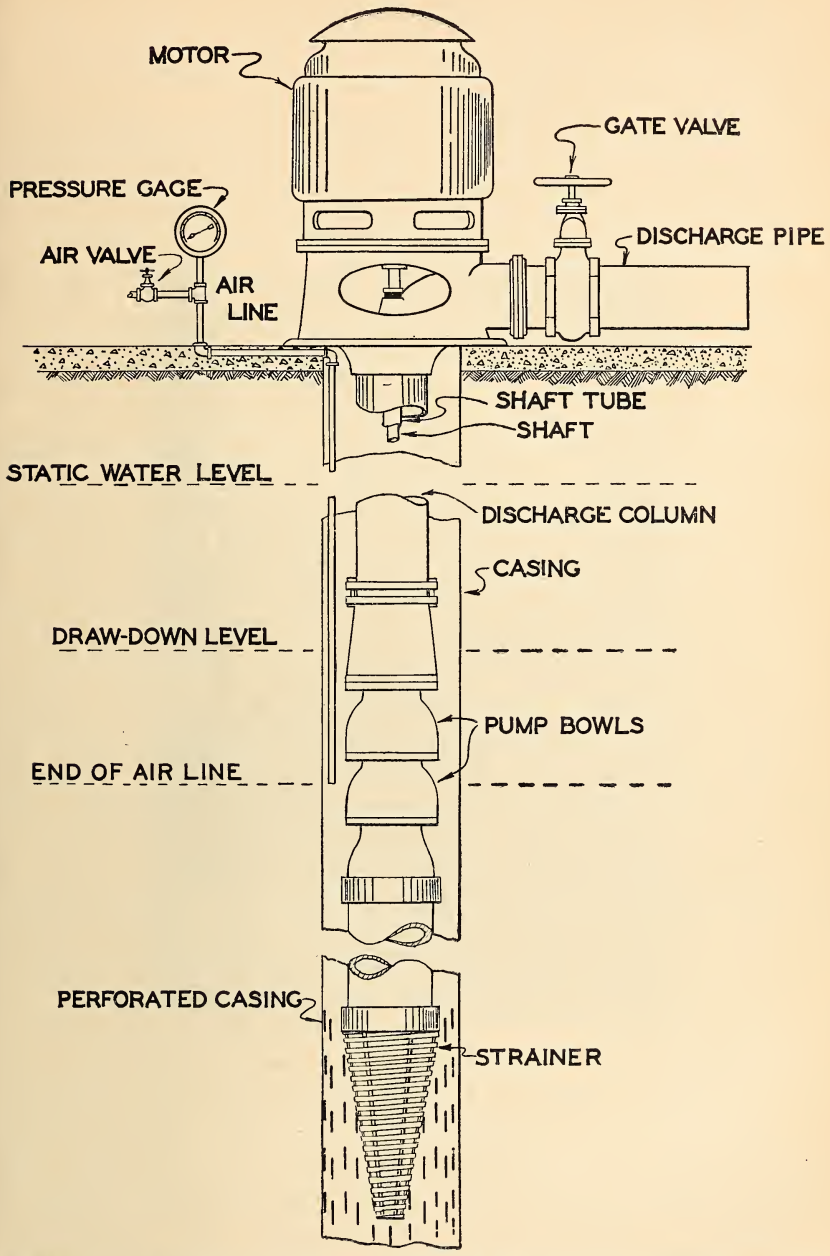


FIGURE 27.—Turbine-pump installation showing air lines and pressure gage for determining distance to water. Pump for building up pressure in air line is not shown.

If the pump does not have an air line, an electric sounder should be used. This sounder makes an electrical contact when the end of the line strikes the water surface. The several types of electric sounders are described in another bulletin (15). The distance to water is ascertained by measuring the length of line in the well or by direct readings, if the line is graduated.

The head on the discharge side of the pump above the center line of the discharge elbow is measured in the same manner as the head on a centrifugal pump. The velocity head is not considered as part of the pumping head in deep-well turbines; therefore, no correction is made for the head required to give the water its velocity.



FIGURE 28.—Hoff meter being used in measuring discharge from pump.

When testing a pumping plant the quantity pumped must be accurately measured. The method used will depend on the equipment available and the conditions existing at the plant. If the discharge is into an open ditch, a weir, a Parshall flume, or a pipe orifice may be used. The methods of making discharge measurements by means of a weir or a Parshall flume are described in *Farmers' Bulletin 1683, Measuring Water in Irrigation Channels*, and other bulletins (2). The discharge tables for weirs and Parshall flumes are based on calibrations of these devices having the dimensions set out in the bulletins, and in order to obtain accurate results the dimensions must be closely adhered to. Current meters may be used to measure

large flows in open channels, but they are not satisfactory for measuring small flows. Propeller-type current meters may be used to measure the pump discharge in the outlet pipe, as shown in figure 28.

Pipe orifices consist of calibrated apertures in fittings which are screwed on the end of the discharge pipe. The discharge depends on the diameter of the pipe, the diameter of the orifice, and the head. It is important that calibrated orifices be used and that the head be measured in the same manner as that followed when the orifice was calibrated. The size of orifice should not be more than 75 percent of the pipe diameter, and it should be such that the head produced will be not less than two or three times the pipe diameter. The additional head caused by the orifice should be included in the total pumping head when the power requirements are checked.

If the pump discharges into a closed line, special methods of measurement must be adopted. There are several usable types of flow meters operating on the Pitot tube principle. They consist of a tube, with one or more impact and pressure openings, which is inserted in the pipe at a convenient point through a packing gland. Through small pipes inside the tube the pressure head and combined impact and pressure heads are transferred to a differential manometer which records the head due to impact. A sliding scale calibrated in feet per second, which gives a direct reading of the velocity, is used to measure the pressure differential. In order to obtain a true average of the velocity in the pipe, readings should be taken at definite intervals along the pipe diameter. From this information the area to which each velocity applies can be determined. By summing up the products of the individual areas and velocities, the total discharge can be determined. In order to obtain satisfactory results when using devices of this type, it is important that they be accurately calibrated and that a sufficient number of readings be taken to disclose all the important changes in the velocity. It is important also that the inside diameter of the pipe be known accurately. Readings with this type of gage should be taken as far as possible from elbows, valves or other fittings that might cause serious irregularities in the distribution of velocity in the pipe.

The color method and the salt-solution method also may be used on closed pipe lines. The velocity is determined by the rate at which the color or salt solution passes through the pipe. The color or the salt solution is injected under pressure at one point and the time taken in reaching another point not less than 100 feet away is determined by means of a stop watch or by electrical means. The time at which the color reaches the second point in the discharge line is observed visually, but when a salt solution is used some method of measuring the change in electrical resistance has to be adopted. Since the velocity of the water in the pipe is a function of the pipe diameter as well as the quantity of water, it is important that the inside diameter be accurately measured. If the pipe is encrusted with rust or partially filled with deposits of sand or silt, this method will not yield accurate results.

The Venturi meter is an accurate measuring device which may be used to determine the flow in pipe lines if there are flanged or flexible couplings in the line where a length of pipe may be removed for inserting the Venturi meter. Since the coefficient of discharge of the Venturi meter varies between 0.94 and 0.98, the values determined from

a calibration of the meter should enter into the calculations if accurate results are to be obtained. When using a Venturi meter of different diameter than the discharge pipe, a sufficient length of pipe of the same diameter as the inlet to the meter should be installed upstream from the meter to allow the flow to become uniformly distributed before it reaches the gage connections. The Venturi meter is not satisfactory for measuring low velocities.

Whichever method of determining the discharge is adopted, enough readings should be taken to get a true average of the discharge. Measurements should be made by two different methods where feasible, so that one can be used as a check against the other.

For measuring the power consumed by electrically driven units, the watt-hour meter at the plant is generally used unless the meter is some distance from the motor. Where this occurs an indicating watt-hour meter or a voltmeter and ammeter are used to measure the power consumption, or a correction is made to the watt-hour readings to take care of the line loss. If the voltage is more than 10 percent above or below normal, voltmeter and ammeter readings should be taken because these abnormalities affect the efficiency of the motor and must be taken into consideration when an acceptance test is being made. When no instrument transformers are used, the horsepower input to the motor is obtained by multiplying the revolutions per second of the meter disk by the horsepower constant of the watt-hour meter. If instrument transformers are used the results should be multiplied by the product of the current transformer ratio and the potential transformer ratio. If no instrument transformers are used, the kilowatt consumption is equal to 3.6 times the product of the revolutions per second of the meter disk by the meter constant stamped on the meter or meter disk. If instrument transformers are used, this product must be multiplied by the transformer ratios in the same manner as that followed when horsepower consumption is being computed. It is usually desirable to have a representative of the power company check the electrical readings, especially when there are unusual conditions or when indicating watt-hour meters, ammeters, and voltmeters have to be used.

The power produced by engine-driven plants cannot be determined by ordinary field tests. However, if the plant guarantee is based on the consumption of a certain quantity of fuel when an acre-foot of water is being pumped against a definite head, the quantity of fuel actually consumed can be accurately determined by weighing or measuring the fuel burned in a definite time, usually an hour or more. The number of acre-feet of water pumped is obtained by multiplying the discharge in cubic feet per second by the length of the test in seconds and dividing by 43.560. The pumping head is measured in the manner previously explained. Since the fuel consumption is directly proportional to the head and the discharge within the ordinary range of differences between the observed head and discharge and the head and discharge specified in the guarantee, it is possible by simple proportion to compute the fuel consumption under the conditions of the guarantee from the observed data.

For example, the guarantee of an engine and pump specified a fuel consumption of 18 gallons of gasoline per acre-foot pumped against a head of 60.0 feet and the test shows that the fuel consumption is 3.1

gallons in 60 minutes when 2.15 cubic feet per second is being pumped against a head of 59.0 feet. As previously explained, the amount pumped will be

$$\frac{2.15 \times 60 \times 60}{43,560} = 0.178 \text{ acre-feet}$$

Since the observed head is 59.0 feet instead of 60, the fuel consumption for pumping 0.178 acre-foot against a head of 60.0 feet will be

$$3.1 \times \frac{60}{59} = 3.15 \text{ gallons}$$

The fuel consumption per acre-foot will be

$$3.15 \times \frac{1}{0.178} = 17.7 \text{ gallons}$$

which is very close to the guaranteed fuel consumption.

The observed consumption of fuel or electrical energy rarely agrees exactly with the guarantee, but this is to be expected because neither the fuel (or energy consumption) nor the head (or discharge) can be accurately determined in the field. It is for this reason that a 5 percent variation is permitted under the terms of the purchase agreement.

In every pump test the speed of the pump should be carefully determined, because the head, discharge, and power consumption are all affected by the speed. (See pages 13 to 22.) If the test shows that the pump does not come up to specifications, it may be that it is not operating at its proper speed. This condition is most likely to occur in engine-driven plants, although it may occur in direct-connected motor-driven plants if the motor is overloaded or if the voltage is low. The speed is usually determined by means of a revolution counter and stop watch, but special tachometers which give direct readings of the revolutions per minute are also used.

Most power companies that have a large agricultural pumping load maintain test crews who make tests without charge. They do not ordinarily make acceptance tests. A pumping plant owner should avail himself of this service from time to time, especially if he has reason to suspect that there has been an appreciable decrease in the efficiency or capacity of his plant. If the results of the acceptance tests are available, any changes that may have occurred will be readily apparent.

COST OF PUMPING FOR IRRIGATION

The cost of pumping consists of the fixed charges and the cost of operation. The fixed charges depend on the cost of the plant and the rate of interest, depreciation, and taxes. These charges are practically independent of the quantity of water pumped or the length of time the plant is operated each year; they have to be met even when the plant is idle. It is for this reason that they are known as the fixed charges. Since they have to be considered in figuring the annual cost of the water pumped, it is evident that the portion of this cost that has to be charged to each acre-foot will decrease as the total quantity pumped increases.

On the other hand, the cost of operation (that is, the expenditure for fuel, electric energy, lubricating oil, repairs, and attendance) is directly proportional to the quantity of water pumped and is zero when the plant is idle. For these reasons the investment in the plant must be kept to a minimum when it is to be operated a short time each year and only a small quantity of water is to be pumped, whereas a large expenditure is permissible in order to obtain the most efficient plant when the pump is to be operated throughout the entire season and a correspondingly large quantity of water is to be pumped.

In preparing an estimate of the cost of pumping when investigating the feasibility of a proposed pumping plant, the prospective purchaser will find that data based on records from existing plants having modern equipment will give more reliable results than data from laboratory tests or theoretical analyses. Table 16 shows the fuel or energy consumption per acre-foot per foot of lift of representative plants of recent construction having modern equipment and high efficiency. By multiplying the average of the values given in the table for the particular fuel by the total head at the proposed plant the quantity of fuel required to pump an acre-foot is obtained. The cost of fuel can then be readily obtained by multiplying by the unit cost of the fuel in the area.

To the cost of fuel must be added the cost of lubricating oil and grease, repairs, and attendance in order to get the operating cost. The cost of oil and grease for an electric plant will be negligible; for a plant using natural gas, it will be about one-tenth the cost of the fuel, and for a Diesel plant it will be about one-third the cost of the fuel. For a gasoline plant the cost of lubricating oil and grease will be between the corresponding costs for natural gas and Diesel plants. However, considerable variation is to be expected in the amount of lubricating oil used because the condition of the engine has a marked effect on the oil consumption.

TABLE 16.—*Energy consumption of small modern irrigation pumping plants per acre-foot per foot of lift*¹

| Type of pump | Type of energy | Year | State | Number of plants | Energy consumption per acre-foot per foot of lift | | | |
|------------------------|----------------|------|------------|------------------|---|---------|---------|---------|
| | | | | | Maximum | Minimum | Average | Unit |
| Deep-well turbine | Natural gas | 1938 | Oklahoma | 1 | ----- | ----- | 36.8 | Cu. ft. |
| | | 1939 | California | 1 | ----- | ----- | 29.1 | Cu. ft. |
| | | | Kansas | 9 | 38.4 | 24.8 | 31.1 | Cu. ft. |
| | Gasoline | 1939 | do | 12 | .41 | .20 | .31 | Gal. |
| | Distillate | 1936 | Colorado | 1 | ----- | ----- | .25 | Gal. |
| | | 1938 | Kansas | 2 | .28 | .27 | .27 | Gal. |
| | Diesel fuel | 1936 | Colorado | 4 | .20 | .11 | .15 | Gal. |
| | | 1935 | do | 1 | ----- | ----- | 1.56 | Kw.-hr. |
| | Electricity | 1938 | California | 7 | 1.64 | 1.47 | 1.58 | Kw.-hr. |
| | | | Idaho | 15 | 2.37 | 1.50 | 1.86 | Kw.-hr. |
| Horizontal centrifugal | do | 1939 | Kansas | 16 | 2.20 | 1.55 | 1.85 | Kw.-hr. |
| | | | do | 4 | 1.76 | 1.67 | 1.72 | Kw.-hr. |

¹ Compiled from the following:

LANGHAM, W., and McMILLEN, W. N. DEEP WELL IRRIGATION IN THE OKLAHOMA PANHANDLE. [Okla.] Panhandle Agr. Expt. Sta. Bul. 64, 22 pp., illus. 1939.

McCALL, K. D., and DAVIDSON, M. H. COST OF PUMPING FOR IRRIGATION. Kans. State Bd. Agr. Rpt. (1939) 58 (234): 1-55, illus. 1939.

Unpublished data from M. R. Kulp of the Idaho Agr. Expt. Sta., W. E. Code of the Colo. Agr. Expt. Sta., The Peerless Pump Co., and the Highline Mutual Water Co. of Calif.

Electric plants require only a nominal amount of attendance to care for them properly. It is estimated that 1 hour for each 24 hours of operation will be sufficient. Much closer attention must be given to Diesel plants to keep them properly lubricated, and they should be visited at regular intervals to see that they are functioning properly; otherwise serious injury to the equipment may result. A safe estimate is to allow about 1 hour for attendance for each 10 hours of operation. Gasoline and natural gas engines require about the same amount of attention as Diesel engines.

The fund that should be set aside annually to take care of repairs is difficult to estimate because a new pump and engine ordinarily require no repairs for several years, but as the age of the equipment increases more and more parts have to be replaced. However, the cost of repairs should be distributed equally throughout the life of the equipment; therefore a lump sum is usually added each year to the operating cost for repairs. Since the money that must be expended for repairs on a specific plant bears a fairly definite relation to the amount the plant is used, a reasonable estimate of the allowance for repairs can be made on the basis of the cost of fuel used, because fuel consumption is directly proportional to the hours of operation.

For a plant driven by natural gas a reasonable estimate of the cost of repairs is from 15 to 25 percent of the cost of the fuel consumed. The actual cost of repairs on gasoline-driven plants is about the same, but since the fuel cost is nearly three times as much for gasoline-driven plants, an allowance of from 5 to 10 percent of the fuel cost is more nearly correct for gasoline plants. For Diesel engines the fuel cost is low, and a larger percentage must therefore be set aside for repairs. From 20 to 25 percent of the fuel cost should be sufficient. If an electric motor is given reasonable care there should be practically no repairs necessary throughout its life. The only repairs necessary on an electric plant would be those on the pump. On the average these should not exceed 5 percent of the cost of the electricity consumed.

Fixed charges consist of the interest on the money invested in the plant, the taxes, and the depreciation. Under present conditions 5 percent may be taken as a reasonable rate of interest. Taxes will amount to from 1 to 2 percent, depending somewhat upon the age and condition of the plant. The depreciation on the various items comprising the plant depends on the expected life of these items. With ordinary care, an electric motor should last from 15 to 20 years, depending to some extent on the amount of use each year. The life of a gasoline or natural-gas engine may be taken as 7 to 10 years, and the life of a Diesel engine as 8 to 12 years, depending on the hours of use each year. If properly cared for and properly loaded, an engine will give a fairly definite number of hours of use, and the life of an engine is, therefore, roughly equal to the total hours of life of the engine divided by the hours of use each year. The life of the pump itself, except under adverse conditions, is from 10 to 15 years, and a conservative estimate of the life is 12 years. The life of a well varies widely in different parts of the country. It may be anywhere between 10 and

30 or more years. Where irrigation pumping has been practiced for a considerable time, information is usually available as to the life of wells, but where sufficient time has not elapsed to determine the life of wells or where there are no wells in existence, a life from 15 to 20 years may be reasonably assumed. Wells deteriorate when not in use, and for that reason it is advisable to operate the pump at least a short period each year to wash accumulated rust from the perforations. The life of the pump house is from 30 to 40 years.

In computing depreciation, a simple method is to charge off each year an amount equal to the total cost of the item divided by the life expectancy. The depreciation obtained in this manner is too high; a more exact method is to determine the amount which with interest compounded semiannually at the standard rate will provide a sum equal to the cost of the item in a period equal to its life. However, since the life of each part of the plant is not known within wide limits these refinements in methods of computing the depreciation do not seem warranted in making an estimate of the fixed charges for a proposed plant.

Table 17 gives data on the fixed charges and the operating costs for representative plants in various parts of the country. Plant cost data are also included. In this table depreciation is computed on an unamortized basis; that is, the depreciation is taken as the cost of the item divided by its life. The unit costs in the table are based on the hours of operation indicated for each plant. For longer periods of operation, that portion of the cost due to fixed charges would be reduced in proportion to the time, and for a shorter time would be increased in like proportion. The values given may be used for the purpose of estimating costs because they are based on results obtained from well-designed plants operating through sufficient periods so that the fixed charges are not out of proportion to the operating costs.

The effect on the costs reflected by pumping different quantities of water per season is clearly shown in table 18 which is based on a theoretical study of electric, Diesel, gasoline, and natural gas plants with a capacity of 900 g. p. m. against a total head of 50 feet (18). Plant costs are also given. The greatest reduction in total cost per acre-foot per foot of lift appears to occur when the quantity pumped is increased from 100 to 200 acre-feet per season. This corresponds to increasing the time of operation from 600 to 1,200 hours per season. Further increases in the quantity pumped or in the time of operation reduce the cost but not to such a great extent as in the first case.

Studies of the cost of pumping for irrigation made 15 to 20 years ago showed unit costs considerably in excess of the values given in tables 17 and 18. The reduction in cost is due in part to the lower cost of fuel and electrical energy, but mostly to the increase in efficiency in pumps. Recognition of the effect of the quantity pumped on the cost has also had a pronounced effect in reducing pumping costs. It is believed that the values given in these tables may safely be used as a basis for studying the economic feasibility of a proposed plant.

TABLE 17.—Operating costs and total cost of pumping 1 acre-foot 1 foot with modern efficient plants using different kinds of fuel or energy ¹

| Year | Type of plant | Total cost of plant | Hours of operation | Quantity pumped | Plant efficiency | Lift | Fuel or energy consumption per acre-foot | Operating cost | | Total cost | |
|------|-----------------------------------|--|--|--|--------------------------------------|--------------------------------------|---|---|---|---|---|
| | | | | | | | | Per acre-foot | Per acre-foot | Per acre-foot | Per acre-foot |
| 1936 | Diesel | { 4, 056 3, 505 4, 640 4, 210 } | { 2, 583 2, 984 1, 940 1, 456 } | { 494 480 438 284 } | { --- --- --- --- } | { 86 91 77 66 } | { Gal. 0.13 .17 .11 .20 Cubic feet 32.7 36.8 } | { Dollars 1.17 2.07 .80 1.41 } | { Dollars 0.014 .023 .010 .021 } | { Dollars 2.70 3.18 3.56 3.71 } | { Dollars 0.031 .035 .031 .056 } |
| | | | | | | | | | | | |
| | | | | | | | | | | | |
| | | | | | | | | | | | |
| 1938 | { Natural gas Gasoline } | { 3, 850 5, 911 3, 000 } | { 2, 170 1, 719 2, 250 } | { 520 273 2, 500 } | { --- --- --- } | { 87 163 85 } | { Kw.-hr. 1.89 1.69 2.2 } | { Dollars 1.19 2.40 2.20 } | { Dollars .014 .015 .026 } | { Dollars 1.94 4.69 2.97 } | { Dollars .022 .029 .035 } |
| | | | | | | | | | | | |
| 1939 | Electric | { 3, 296 3, 419 3, 056 3, 207 3, 406 } | { 3, 104 2, 920 2, 250 3, 135 3, 070 } | { 380 613 334 603 1, 126 } | { 57 60 62 64 68 } | { 36 36 36 36 33 } | { Kw.-hr. 1.89 1.69 1.62 1.59 1.50 } | { Dollars .933 .813 .871 .811 .639 } | { Dollars .026 .023 .024 .022 .020 } | { Dollars 2.15 1.59 2.15 1.56 1.06 } | { Dollars .059 .044 .060 .043 .033 } |
| | | | | | | | | | | | |
| | | | | | | | | | | | |
| | | | | | | | | | | | |

¹ Compiled from the following:
 LANGHAM, W., and McMILLEN, W. N., DEEP WELL IRRIGATION IN THE OKLAHOMA PANTHANDLE, [Okla.] Panhandle Agr. Expt. Sta. Bul. 64, 22 pp., illus. 1939.
 McCALL, K. D., and DAVISON, M. H., COST OF PUMPING FOR IRRIGATION. Kans. State Bd. Agr. Rpt. (1939) 38 (234): 1-55, illus. 1939.

Unpublished data from M. R. Kulp of the Idaho Agr. Expt. Sta., W. E. Code of the Colo. Agr. Expt. Sta., and the Highline Mutual Water Co. of Calif.
 The cost includes interest, depreciation, taxes, fuel or energy, lubricating oil, and repairs, but not attendance.
² Quantity pumped is assumed.

TABLE 18.—*Total cost of pumping 1 acre-foot 1 foot with different kinds of power, for different quantities of water (18)*

| Kind of power | Total cost of plant including well | Unit cost of fuel or energy | Total cost ¹ of pumping 1 acre-foot 1 foot when pumping— | | | |
|------------------|------------------------------------|-----------------------------|---|-------------------------|-------------------------|-------------------------|
| | | | 100 acre-feet per annum | 200 acre-feet per annum | 300 acre-feet per annum | 400 acre-feet per annum |
| | <i>Dollars</i> | <i>Cents</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> | <i>Dollars</i> |
| Electric..... | 1, 606 | 0.02 per kw.-hr..... | 0. 073 | 0. 057 | 0. 051 | 0. 048 |
| Diesel..... | 3, 330 | 0.07 per gal..... | . 098 | . 062 | . 049 | . 043 |
| Gasoline..... | 2, 120 | 0.11 per gal..... | . 085 | . 062 | . 054 | . 050 |
| Natural gas..... | 2, 120 | 0.30 per M cu. ft..... | . 071 | . 048 | . 040 | . 036 |

¹ Includes interest, taxes, insurance, plant depreciation, fuel, plant repairs, plant construction, and attendance. It does not include field distribution of the water.

SUMMARY

The quantity of water that must be supplied to the crop by the pumping plant is the difference between the water requirements of the crop and the rainfall. To this must be added the quantity necessary to take care of the losses incurred in getting the water to the crop. If some water is available from other sources, the quantity pumped will be reduced by this amount. The water requirements of crops range from 18 to 42 inches and of this amount from 20 to 40 percent will be required during the month of peak demand. The capacity of the pumping plant must be sufficient to take care of the needs during this period of greatest use.

The area that can be irrigated by the pumping plant depends on the capacity of the well or the surface supply from which the water is drawn. For a project to be successful, the pumping lift must not exceed the economic limit, and if the water is drawn from a well, the draft must not exceed the ground-water supply. During periods of shortage a temporary overdraft is permissible if the ground-water supply will be replenished after the emergency is over.

The economic limit of lift depends on the cost of fuel or power, the crops grown, the climate, the productiveness of the soil, the method of irrigation, and the ability of the farmer. In most areas the economic limit of lift has been determined by experience. Where it is necessary to exceed this limit all the factors involved should be carefully investigated in order to find out whether exceeding the recommended lift is justified.

The discharge of a well is a function of the draw-down. The discharge per foot of draw-down for wells in thin water-bearing formations usually decreases as the draw-down increases, whereas it remains practically constant for artesian wells and deep wells in thick water-bearing formations. The relation between draw-down and discharge is determined by a well test, which is made by pumping the well at various rates and noting the draw-down and discharge for each rate. This information is necessary when the pump is being purchased.

The horizontal and vertical centrifugal and the deep-well turbine are the types of pumps most generally used in pumping water for irrigation. The horizontal centrifugal is best suited for pumping from surface water supplies where the fluctuation in the water surface is small, and for pumping from wells where the water table is less than 20 feet beneath the surface. Where the water level fluctuates widely a

vertical centrifugal or deep-well turbine should be used. The deep-well turbine is the best type of pump for wells where the depth to water is more than 20 feet, but pumps of this type are frequently used in shallow wells because they do not require priming.

Deep-well turbines are of three types depending on the kind of impeller used. They are the true deep-well turbine which operates on the centrifugal principle; the mixed-flow turbine in which the action of the impeller is a combination of centrifugal force and direct thrust; and the axial flow or screw pump in which the impeller is of the screw or propeller type that raises the water by direct thrust of the blades of the impeller. Centrifugal impellers produce a high head per stage but have small capacity. Axial-flow impellers have small lifting power per stage, but large capacity. Mixed-flow impellers have characteristics between those of the centrifugal and axial-flow types.

The characteristics of horizontal centrifugal pumps and deep-well turbines are shown by the performance curves. These curves show the efficiency, total head, and brake horsepower when different quantities of water are being pumped. From the performance curves it is possible to choose the pump best suited to a specific pumping problem. To obtain the highest efficiency from a pump, it should be operated at the capacity for which the efficiency is at its peak, as shown by the efficiency curve; and to avoid overloading the power plant if the water level changes, the horsepower and efficiency curves should have their peaks at the same discharge.

The load, the capacity, and the horsepower of a centrifugal pump are functions of the speed and the size of the impeller. For small changes in speed, the discharge is proportional to the speed, the head is proportional to the square of the speed, and the horsepower is proportional to the cube of the speed. The same relations hold with reference to the diameter of the impeller.

The efficiency of the pump is the ratio of the water horsepower produced to the horsepower required to drive the pump. The over-all efficiency is the ratio of the water horsepower produced by the pump to the total power input of the plant.

If a pump delivers too much or too little water, it can be made to fit the required conditions by changing the speed of the pump or the diameter of the impeller. Belt or engine-driven plants can be made to fit the required conditions most easily by changing the pump speed, but direct-connected motor-driven plants, if they are to operate efficiently, can be made to fit the conditions only by changing the diameter of the impeller, since the speed of the motor is fixed.

The electric motor is the most dependable source of power for pumping and in the long run where electric rates are reasonable it is probably the cheapest source of power. The three-phase, alternating-current induction motor, because of its low cost, long life, high efficiency, and dependability, is most frequently used to drive irrigation pumps. When equipped with ball bearings it can be run in either the horizontal or the vertical position.

Most electric rates contain a demand charge based on the horsepower of the motor. For this reason, it is most economical to choose the smallest plant that will deliver the required quantity of water when needed in a stream of sufficient size to be applied to the crops efficiently.

Internal-combustion engines are used as a source of power for pumps in areas where electricity is not available or where the rates are too high. They are of two types, electric-ignition engines and Diesel engines. Internal-combustion engines suitable for the different types of fuel are available in sizes to fit practically every irrigation need. The fuels used at present in internal-combustion engines are natural gas, liquid gas (butane), gasoline, kerosene, distillate or tractor fuel, and fuel oil. The lighter fuels are adapted for use in electric-ignition engines, but fuel oil can be used only in Diesel engines. The choice of the type of engine for a plant will depend on the life, efficiency and the cost of the engine, the type of fuel available for use, the relative cost of the different fuels, and the hours of operation of the plant. The engine that makes the total cost of pumping the least should be chosen.

Automobile engines may be used to drive pumps, but they should not be loaded to over 50 to 75 percent of their rated power, depending on their condition. Heavy-duty electric-ignition and Diesel engines should not be loaded to over 75 to 80 percent of their rated power.

To get the best service from engines, they should be kept in good condition and should be provided with protective devices such as oil and fuel filters, air cleaners, and automatic shut-offs. The fuel should be of the proper grade, and particular attention should be given to the lubricating oil.

The types of drives most often used in transmitting power from the engine or motor to the pump are the direct drive, the gear drive, and the flat or V-belt drive. Of these, the direct drive is most efficient and the flat-belt drive is usually least efficient. Both flat and V belts may be used for quarter-turn drives. Small pulleys lower the efficiency of belt drives and should therefore be avoided.

Riveted or welded sheet-steel pipe, either plain or galvanized and reconditioned standard pipe, and well casing are suitable for pump discharge lines. Concrete pipe or sewer tile may be used if the pressure on the pipe is small and if there is no danger from water hammer or other shocks. Standard pipe or well casing should be used for suction lines because it lasts longer and is easier to keep airtight.

The carrying capacity of pipe is a function of the diameter, the head available, and the friction. The friction increases at a high rate as the velocity in the pipe increases. For this reason, the size of pipe should be carefully proportioned to the quantity of water to be carried. Discharge lines should be laid so that no part of the line is above the hydraulic gradient, and they should have air-relief pipes or valves where pronounced changes in grade occur.

A pitcher pump or a double-acting suction pump provides a simple means of priming pumps and siphons. The priming pump should be attached to the high point of the pump or the siphon.

Check valves are installed in long discharge lines to prevent the water in the pipe from running back into the well when the pump stops. Foot valves are sometimes recommended for this purpose, but they are generally unsatisfactory because they frequently leak.

Check valves and foot valves cause water hammer in the discharge line when they close. The shock due to water hammer can be materially reduced by installing a surge pipe, an air chamber, or a relief valve in the line near the pump. The surge pipe is the simplest means

of reducing the water hammer, but it cannot be used conveniently in high-pressure lines because the required height of the pipe is too great. Under these conditions an air chamber or relief valve is installed. The height of surge pipe and the size of air chamber can be determined by the formulas given in this report.

Alfalfa valves are a cheap and effective means for providing turn-outs on pipe lines where the head is less than 50 feet.

Before purchasing the equipment for a pumping plant, bids should be obtained from several manufacturers because a better price is usually obtained by this method. It is not necessary to call for formal bids unless the investment in the plant is large. The bidders should be supplied with accurate information as to the depth to the static water level, the quantity of water to be pumped, the drawn-down, the fluctuation in the water level, the diameter of the well, and the kind of power available. After the farmer has decided on the make of pump and other equipment he plans to buy, a purchase agreement with the pump manufacturer should be entered into. The purchase agreement should specify and describe the equipment to be furnished and all details of the agreement. The most important of these is the performance guarantee, which is a statement of the power required to pump a specified quantity of water against a specified head. It is generally more satisfactory to have one contract covering all the equipment needed for the plant than to have separate agreements for the different items, such as pump and motor or engine.

A test of the equipment should always be made after the plant is completed to see whether it meets the terms of the purchase agreement. This test consists of the determination of the head, the discharge, and the power consumed. In making the tests a deviation of as much as 5 percent from the terms of the contract is allowable because of unavoidable errors in field tests. It is difficult to determine the horsepower of engine-driven plants in the field and for this reason it is customary in the purchase agreement, to specify the fuel consumption rather than the horsepower.

The cost of pumping is made up of two items, fixed charges and operating costs. The fixed charges (interest, depreciation, and taxes) accrue whether the plant is used or not. The operating costs (power charges, lubricating oil, attendance, and repairs) are proportional to the quantity of water pumped. For this reason, the total cost of pumping a unit quantity of water decreases as the total quantity pumped increases, but at a decreasing rate.

Tests indicate that well-designed, efficient, modern pumping plants using the different types of fuel require about 35 cubic feet of natural gas, one-third of a gallon of gasoline, one-fourth of a gallon of distillate, one-seventh of a gallon of Diesel fuel, or 1.7 kilowatt-hours of electrical energy to lift 1 acre-foot 1 foot. With this type of plant when operating 2,000 hours or more each season, the total cost of pumping (fixed charges and operating cost exclusive of attendance) is between \$0.03 and \$0.05 per acre-foot per foot of lift. Although these costs are based on a limited number of tests and depend to a considerable extent on the cost of the fuel or energy, they show what can be expected from efficient plants under ordinary conditions as to prices. A slightly higher pumping cost when electricity is used is generally considered justifiable because of the convenience and the saving in cost of attendance.

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